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THERMAL EXPANSION STRESSES IN PIPING.

By MASON S. NOYES, Member.*

The author presents, with all necessary information and instructions for its application, the established method used by the Bureau of Ships for designing piping to take care of thermal expansion stresses. The method is here published for the first time. It was originated by Mr. H. C. E. Meyer, is based on previous methods presented by Professor Wm. Hovgaard and hence is designated the "Meyer-Hovgaard" method.

DESIGNATION AND SCOPE.

Relation to Prior Work on same Subject—The system of stress analysis hereby published for the first time has been in continuous and satisfactory use since its inception in 1934. In application it

* Bureau of Ships, Navy Department.

has been extended beyond the simple shapes of piping section covered herein to the solution of complex single sections and of many-branched sections. In order to keep the present article within reasonable limits, complex and branched sections of piping are left for later consideration.

The system herein described has been prepared for the purpose of offering, and in use it has demonstrated, certain practical advantages not possessed by the Walker-Crocker method (I)* (II) (III) (IV), the Hovgaard method (V) (VI) (VII), and the Karelitz-Marchant method (VIII). Testimony in part to this fact is found in the last paragraph of Bleick's discussion (IX) and in Hovgaard's discussion (VIII, March 1938, page A-37).

Description of the system necessarily must assume that the reader already is familiar with the above references. Successful application of the system is dependent only on careful compliance with the instructions given hereinafter for the use of the accompanying forms.

Basis for Present System—The system herein described is a simply formulated bookkeeping of the analytical method set up by Hovgaard (V) (VI) (VII). By means of it any person capable of accurate arithmetic but unacquainted with the theory underlying it can perform the necessary calculations.

The system, as originated by H. C. E. Meyer† with characteristic thoroughness and clarity, is arranged in a series of integrated steps based upon independent consideration first of each reference plane XY, YZ and ZX at one anchorage and second of each bent and straight element in the section of piping relative to each reference plane. It may be considered as equivalent to working simultaneously, the two Walker-Crocker (I) diagrams for each plane, considering each element completely singly and consecutively from anchorage to far end, then summing the elements horizontally, combining lastly above and below the base line to get values determinant of the forces and moments in that particular plane.

The actual steps in the system are controlled and automatically coordinated by the use of forms which cover each part of each step and which comprise finally a complete record of the calculations, ready for check-up and future reference.

* Roman numerals refer to references listed in the bibliography.

† Chief Engineer, Gibbs & Cox, Inc., N. Y. C.

Designation of Present System—The system herein described is based as stated on the method developed by Hovgaard (V) (VI) (VII) for the determination of the forces and stresses caused by thermal expansion. Therefore the system will be identified hereafter as the Meyer-Hovgaard method.

Description.

Hovgaard Method—On the assumption of the reader's familiarity with the method developed by Hovgaard (V) (VI) (VII), no detailed description of the mathematics and tests on which the method is based is included herein.

One fundamental assumption is made, however:

When a given section of piping connecting two fixed points expands in length due to thermal rise, forces and moments will be produced at the fixed points which are exactly equal to the forces and moments which would be required at these fixed points to force the unrestrained section of piping after the same thermal rise back to its original direction at the fixed points.

Thus if the single-plane section of piping between the two fixed points "O" and "A" in Figure 1 undergoes a rise in temperature,

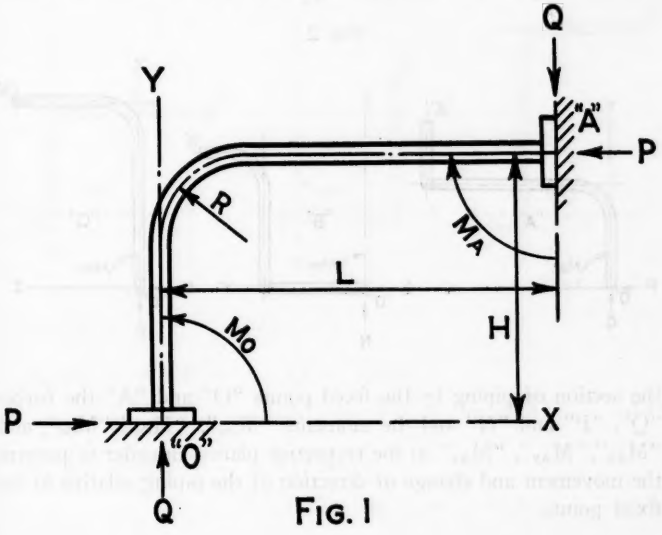


FIG. 1

there will be imposed on the section of piping by the fixed points "O" and "A" the forces "Q" and "P" and the moments "M_O" and "M_A" in order to prevent the movement and change of direction of the piping at the fixed points.

Similarly if the section of piping lies in three planes as in Figure 2 and undergoes a rise in temperature, there will be imposed on

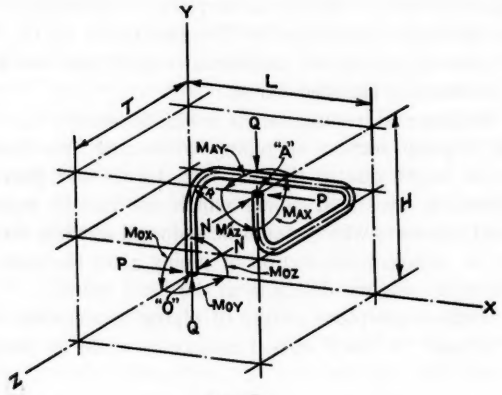
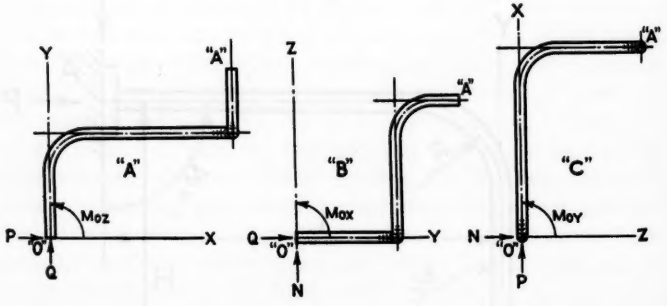


FIG. 2



the section of piping by the fixed points "O" and "A" the forces "Q", "P" and "N" and the moments "M_{Ox}", "M_{Oy}", "M_{Oz}", and "M_{Ax}", "M_{Ay}", "M_{Az}", in the respective planes, in order to prevent the movement and change of direction of the piping relative to the fixed points.

Summary of Hovgaard Formulae—The fundamental formulae for a section of piping between fixed points are given by Hovgaard as follows:

For a single plane, Figure 1, by (V) (second paper, sections B-4 and B-7):

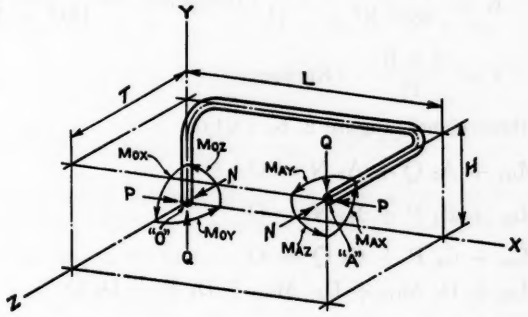
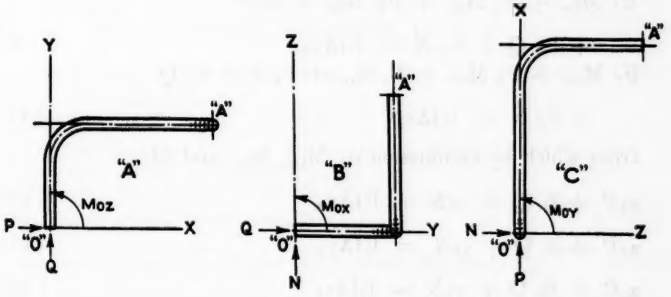


FIG. 3



$$A_s M_o + A_y P - A_x Q = O \qquad (1)$$

$$A_y M_o + A_y^2 P - A_{xy} Q = EI\mu L$$
$$= EI\mu L \text{ for Fig. 1} = EI\Delta x \qquad (2)$$

$$A_x M_o + A_{xy} P - A_x^2 Q = O$$
$$= -EI\mu H \text{ for Fig. 1} = -EI\Delta y \qquad (3)$$

where $A_S = \int_o^A K ds \quad A_y = \int_o^A K y ds \quad A_x = \int_o^A K x ds$

$$A_y^2 = \int_o^A K y^2 ds \quad A_{xy} = \int_o^A K x y ds \quad A_x^2 = \int_o^A K x^2 ds$$

$$K = \frac{48h^2 R^2 + 10r^4}{48h^2 R^2 + r^4} \text{ (Hovgaard)} = \frac{12\lambda^2 + 10}{12\lambda^2 + 1} \quad (4)$$

$$\lambda = \frac{2 h R}{r^2} \quad \text{(Karman)} \quad (5)$$

For three planes, Figure 2, by (VI):

$$A_1 M_{Ox} + A_5 Q + A_6 N = O \quad (6)$$

$$B_1 M_{Oy} + B_4 P + B_6 N = O \quad (7)$$

$$C_1 M_{Oz} + C_4 P + C_5 Q = O \quad (8)$$

$$D_1 M_{Ox} + D_2 M_{Oy} + D_3 M_{Oz} + D_4 P + D_5 Q + D_6 N = EI\Delta x_A \quad (9)$$

$$E_1 M_{Ox} + E_2 M_{Oy} + E_3 M_{Oz} + E_4 P + E_5 Q + E_6 N = EI\Delta y_A \quad (10)$$

$$F_1 M_{Ox} + F_2 M_{Oy} + F_3 M_{Oz} + F_4 P + F_5 Q + F_6 N = EI\Delta z_A \quad (11)$$

from which by elimination of M_{Ox}, M_{Oy} and M_{Oz}:

$$\alpha_1 P + \beta_1 Q + \gamma_1 N = EI\Delta x_A \quad (12)$$

$$\alpha_2 P + \beta_2 Q + \gamma_2 N = EI\Delta y_A \quad (13)$$

$$\alpha_3 P + \beta_3 Q + \gamma_3 N = EI\Delta z_A \quad (14)$$

For the stresses, by (VI):

$$M_O = \sqrt{M^2_{Ox} + M^2_{Oz}} \text{ or } \sqrt{M^2_{Oz} + M^2_{Oy}} \text{ or } \sqrt{M^2_{Oy} + M^2_{Ox}} \quad (15)$$

according to the axes of the plane normal to section of piping at the "O" end.

$$p_{1b} = \frac{M_O r_o}{I}, \text{ i.e. } \sigma_{Ob} = \frac{M_O r_o}{I} \quad (16)$$

$$p_{1s} = \frac{pr_1}{2t} \text{ , i.e. } \sigma_s = \frac{pr_1}{2t} = \frac{pd_1}{4t} = \tfrac{1}{2}\ \sigma_t \quad (17)$$

$p_{1c} = \frac{Q}{A}$ or $\frac{P}{A}$ or $\frac{N}{A}$ according to which force has the same direction as the section of piping at the "O" end, and normally is negligible, i.e. $\sigma_{Oc} = p_{1c}$ (18)

$$p_1 = p_{1b} + p_{1s} + p_{1c}, \text{ i.e. } \sigma_O = \sigma_{Ob} + \sigma_s + \sigma_{Oc} \quad (19)$$

$$p_2 = \frac{Pr_1}{t} \text{ , i. e. } \sigma_t = \frac{pr_1}{t} = \frac{pd_1}{2t} \quad (20)$$

$$q_t = \frac{Mo_y r_o}{2I} \quad \text{or} \quad \frac{Mo_x r_o}{2I} \quad \text{or} \quad \frac{Mo_z r_o}{2I}$$

according to which moment lies in the plane normal to the section of piping at the "O" end (i.e., the end moment that does *not* determine M_O), i.e., $\tau_t = q_t$ (21)

$$q_f = \frac{\sqrt{P^2 + N^2}}{A} \quad \text{or} \quad \frac{\sqrt{N^2 + Q^2}}{A} \quad \text{or} \quad \frac{\sqrt{P^2 + Q^2}}{A}$$

according to which forces are normal to the section of piping at the "O" end, and normally is negligible, i.e. $\tau_f = q_f$ (22)

$$q = q_t + q_f \text{ , i.e. } \tau = \tau_t + \tau_f \quad (23)$$

$$p_{eq} = \sqrt{p_1^2 + p_2^2 - p_1 p_2 + 3q^2}$$

(Huber-Hencky (V) or strain-energy theory), i.e. $\sigma_{eq} = \sqrt{\sigma_O^2 + \sigma_t^2 - \sigma_O \sigma_t + 3\tau^2} =$ "equivalent stress" (unit resultant). (24)

$$\sigma_{Od} = \sqrt{\sigma_O^2 + \sigma_t^2}, \text{ for single plane sections only.} \quad (24a)$$

The above stress formulae have been given in both the standard (ASA) symbols and the original Hovgaard symbols both to avoid confusion and to facilitate understanding and transition to standard symbols.

For any point "S" (x,y) of a section of piping in a single plane:

$M_S = M_O + Py - Qx$ = moment at the point "S" (x,y). (25)

For any point "S" (x,y,z) of a section of piping in three planes:

$M_{Sz} = M_{Oz} + Py - Qx$ = moment in the XY plane (26)

$$M_{Sx} = M_{Ox} + Qz - Ny = \text{moment in the YZ plane} \quad (27)$$

$$M_{Sy} = M_{Oy} + Nx - Pz = \text{moment in the ZX plane} \quad (28)$$

Nomenclature and Symbols Used—The following definitions are based on the origin of the reference axes being placed at the fixed end designated "O" (Figures 1 and 2):

Section of piping—that portion of a piping system lying between two anchorages or fixed terminals or equipment and consisting of tubing or pipe, fittings, valves, and flanged joints.

Piping—an assembly of pipe(s) or tubing, valves, and fittings forming a whole or a part of a system used for transferring steam, gas, or liquids, and identified for size by I.P.S. designations.*

Pipe(s)—material purchased by I.P.S. designations for size and commonly as "standard", "extra strong", or "double extra strong" for wall thickness; primarily intended for fabrication by pipe threading.*

Tubing (or tubes)—material purchased by outside diameters and wall thicknesses stated in decimals of an inch. This material is intended for fabrication by methods other than pipe threading, such as flanging, rolling, welding.*

Element of a Piping Section—a straight portion lying between two bent portions, or a bent portion having an arc of 90° or less and one or more ends in a plane parallel to or coincident with one of the reference planes XY, YZ, ZX. A straight element sometimes is called a "tangent", with respect to an adjacent bent element.

Direction—direction with respect to a reference axis (X, Y or Z) is taken as positive when it leads away from the origin "O".

The following symbols, corresponding to those standardized by the A. S. A. (X), except as otherwise noted are used:

A = area of the cross-section of the piping material.

A, B, C, D, E, F = coefficients in fundamental formulae, as indicated by a subscript.

α, β, γ = coefficients in transient formulae, as indicated by a subscript.

E = Modulus of elasticity or Young's modulus, taken at operating temperature of piping.

* Quoted with slight modification from the General Specifications for Machinery, Bureau of Ships, Navy Department.

L, H, T = distance of fixed end "A" in the direction of reference axes OX, OY, OZ, respectively.

$h = \frac{1}{2}t$ = half the average wall thickness of the piping material.

I = rectangular moment of inertia (area).

K = flexibility factor, by Hovgaard (see formula (4)).

λ = Karman's coefficient for flattening at a bend (see formula (5)).

M = moment (i.e., couple) acting at the point and about the axis, as indicated by a subscript.

μ = thermal elongation per foot of distance between anchorages (Hovgaard).

N, P, Q = anchorage reactions (i.e., forces) acting in the direction of the Z, X, Y axes respectively.

p = pressure per unit of area (internal).

R = center-line radius of piping at the bend.

$r = h + r_i = \frac{1}{2}(r_o + r_i) = \frac{1}{4}(d_o + d_i)$ = radius of the average middle surface of the piping material cross-section.

$r_o = \frac{1}{2}d_o$ = average outside radius $\}$ of the piping material
$r_i = \frac{1}{2}d_i$ = average inside radius $\}$ cross-section

p = unit normal stress, at the point and of the kind as indicated by a subscript; thus 1 for longitudinal, 2 for transverse (Hovgaard) (see also σ).

q = unit shear stress, at the point and of the kind as indicated by a subscript; (Hovgaard) (see also τ).

σ = unit normal stress, at the point and of the kind as indicated by a subscript; thus b for bending, c for compressive, s for longitudinal (internal pressure), t for transverse (internal pressure) (ASA).

τ = unit shear stress, at the point and of the kind as indicated by a subscript; thus t for twisting, f for direct (ASA).

Δx, Δy, Δz, = total thermal elongation of piping from the origin "O" in the OX, OY, OZ directions respectively, at a point, as indicated by a subscript.

Single Plane Application and Instructions—The fundamental formulae (1),(2),(3) for a piping section in a single plane

already have been given (p. 279). They contain coefficients identified by the letter "A" which are summations of element coefficients based on element developed lengths and relative flexibility. A "flexibility factor" applied to the developed length of each element determines the equivalent effective length to be used in the element coefficients. For straight elements ("tangents") the factor is equal to unity. For bends it supplies the effect of the flattening of the piping cross-section and therefore, as indicated by formulae (4) and (5), depends on the radius of the bend (R), the wall thickness (t = 2h) and the mean radius of the piping cross-section (r). Values of this factor based on formulae (4) and (5) and on the standard dimensions of Navy tubing and A.S.A. pipe are tabulated in Tables III, IV, and V for various ratios of bends. Values for the moment of inertia I are given also.

Knowing the form and dimensions of a given section of piping, the corresponding element coefficients and the resultant summation coefficients "A" can be determined and substituted in the left side of formulae (1),(2),(3). Similarly, knowing the piping material and the working temperature, the proper coefficient of expansion E and the thermal change in overall dimensions caused by the resultant rise in temperature may be substituted in the right side of formulae (1),(2),(3). The three unknowns M_o, P and Q then may be found by solving the formulae as simultaneous equations. Unit stress conditions lastly are investigated by substitution in the stress formulae (16) to (24), inclusive. Usually it is sufficient to determine the bending stress σ_{ob} (16), the transverse tensile stress σ_t (20), and the maximum combined longitudnal stress σ_o (19), and the resultant tensile (diagonal) stress σ_{od} (24a).

The summation coefficients "A" in the formulae (1),(2),(3) although shown as mathematical integrations may be described and determined from the element coefficients as summations on a basis of developed length thus:

A_s = summation of length A_x^2 = summation of x^2

A_x = summation of x A_y^2 = summation of y^2

A_y = summation of y A_{xy} = summation of xy

These summation coefficients can be determined by a number of methods. A graphic form similar to the Walker-Crocker would

consist of setting off as abscissae the element values of x, y, x^2, y^2, and xy opposite the developed length of the corresponding element as an ordinate on a reference axis equal to the total equivalent effective length of the section of piping. The areas under the curves drawn through the successive points can be measured by planimeter and represent the values of the summation coefficients.

As shown by the various methods described by most previous writers on this subject, the summation coefficients may be determined algebraically. Piping sections of simple form and single plane similar to Figure 1 lend themselves very easily to this method of solution. In at least one instance, solution of the most simple and common forms of single plane piping has been reduced to use of a series of curves and charts based on the principal physical dimensions and proportions.

However shipboard piping lines, and often central station mains as well, are complex and more often than not three-plane. Algebraic solution of such complex piping becomes very much involved and susceptible to errors. The Meyer-Hovgaard method performs such a solution in a clear and organized manner. It commences with and, like the graphic method mentioned above, is based upon dividing the section of piping into its elements, determining all the "A" coefficients for each element individually, and establishing the summation coefficients "A" for the entire piping section by summing up these "A" coefficients for all elements.

The formulae for the "A" coefficients for simple elements, such as straight lengths ("tangents") positioned either parallel or oblique to one of the reference axes of a single plane and such as bends of commonly used angles in various positions relative to reference axes of a single plane, have been developed. Each position of a simple element is shown by the Meyer-Hovgaard method on an individual form, together with the corresponding formulae for its "A" coefficients, a sketch identifying the element and its position relative to the reference axes and their origin "O", and a table of its principal dimensions and factors. These forms are numbered for convenient identification. Additional forms are provided for summation of the "A" coefficients taken from the element forms and for the tabulation of the resultant summation coefficients "A", forces P and Q, end moments M_O and M_A, and tensile stresses σ_{Ob}, σ_S, σ_O, σ_t, σ_{Od} and σ_{Ab}, σ_A, σ_{Ad}.

Auxiliary forms also are provided for laying out the dimensions of a single plane piping section, for selecting the applicable element forms, and for tabulation of the form numbers and dimensions applicable to each element.

The end moment M_A is found by substituting in formula (25) the values "L" and "H" (Figure 1) for "x" and "y". The end stresses σ_{Ab}, σ_A, σ_{Ad} are determined thereafter by substituting the corresponding "A" end terms for the "O" end terms in the applicable formulae.

Three Plane Application and Instructions — The fundamental formulae (6) to (14), inclusive, for a piping section in three planes already have been given (p. 280), also. Formulae (6) to (11), inclusive, contain coefficients identified by the letters (A) to (F), inclusive, which are summations of element coefficients based on element developed lengths, relative flexibility, and individual distances from the particular reference axes and plane involved. This last item in effect may be tabulated as follows:

Coefficients	Formula	Axes and Plane
A	6	YZ
B	7	ZX
C	8	XY
D, E, F	9, 10, 11	XY, YZ, ZX

Further coefficients identified by the letters (α), (β), (γ) as combinations of the above coefficients (A) to (F) form a part of formulae (12), (13), (14).

The effect of relative flexibility has been described already under the preceding section.

The right side of formulae (12), (13), (14) or (9), (10), (11) also may be determined easily as described under the preceding section for single plane application.

The left side likewise could be determined in accordance with the preceding description were it not that, as may be seen from Figures 2 and 3, each element now is positioned normal to one of the three reference planes and is subjected to an additional force normal to the plane in which the element is located. The resultant twisting, rotations about reference axes, and displacements not encountered in single-plane piping have been detailed by Hovgaard (VI).

Therefore additional forms are provided under the Meyer-Hovgaard method to cover each position of a simple element in a plane normal to the reference plane.

However, in checking over these added forms for elements in planes normal to the plane of reference, certain minor or secondary terms will be found to have been omitted. These terms purposely were deleted in order to simplify the solution, after it was established by test cases that the resulting difference was not more than the celebrated "one half of one per cent."

A method of stress analysis could be evolved that would include all these neglected secondary terms but it is felt that such a method involves more than triple the work required by the method as now set up. Since the gain from the more accurate method appears to be approximately one-half of one per cent only, the much greater chance for arithmetical error offered by the greatly increased amount of solution made advisable the omission of secondary terms.

Turning now to Figures 2 and 3, it may be noted that the projection on each reference plan of the section of piping has been given and indicated as "A", "B", and "C" for the "XY", "YZ", and "ZX" planes, respectively. From these it is evident that with the forms provided each projection may be considered as a separate single plane and may be solved as if formulae (1), (2), (3) were written:

For the "XY" plane:

$$A_S M_{Oz} + A_y P - A_x Q = 0 \qquad (8a)$$

$$A_y M_{Oz} + A_y^2 P - A_{xy} Q = EI\Delta x_A \qquad (9')$$

$$A_x M_{Oz} + A_{xy} P - A_x^2 Q = -EI\Delta y_A \qquad (10'')$$

For the "YZ" plane:

$$B_S M_{Ox} + B_z Q - B_y N = 0 \qquad (6a)$$

$$B_z M_{Ox} + B_z^2 Q - B_{yz} N = EI\Delta y_B \qquad (10')$$

$$B_y M_{Ox} + B_{yz} Q - B_y^2 N = -EI\Delta z_B \qquad (11'')$$

For the "ZX" plane:

$$C_s M_{0y} + C_x N - C_z P = 0 \qquad (7a)$$

$$C_x M_{0y} + C_x^2 N - C_{zx} P = EI\Delta z_c \qquad (11')$$

$$C_z M_{0y} + C_{zx} N - C_z^2 P = -EI\Delta x_c \qquad (9'')$$

Written thus, it is apparent that the total expansion in the direction of each reference axis is the sum of the same expansion in each of the two planes containing the reference axis. Combining the above like-numbered equations:

$$A_y M_{0z} - C_z M_{0y} + P(A_y^2 + C_z^2) - A_{xy} Q - C_{zx} N = EI\Delta x \qquad (9')—(9'')$$

$$B_z M_{0x} - A_x M_{0z} + Q(B_z^2 + A_x^2) - B_{yz} N - A_{xy} P = EI\Delta y \qquad (10')—(10'')$$

$$C_x M_{0y} - B_y M_{0x} + N(C_x^2 + B_y^2) - C_{zx} P - B_{yz} Q = EI\Delta z \qquad (11')—(11'')$$

Equations (6a),(7a),(8a) give the values of the moments M_{0x}, M_{0y}, M_{0z} in terms of the forces P, Q, N, which values upon substitution in the equations just developed simplify the three total expansions into terms only of the unknown forces P, Q, N:

$$-\beta_1 N + \beta_3 P - \beta_2 Q = EI\Delta x \qquad (12a)$$

$$-\alpha_2 N - \beta_2 P + \lambda_2 Q = EI\Delta y \qquad (13a)$$

$$+\alpha_1 N - \beta_1 P - \alpha_2 Q = EI\Delta z \qquad (14a)$$

where

$$\alpha_1 = C_x^2 + B_y^2 - \left(\frac{B_y^2}{B_s} + \frac{C_x^2}{C_s}\right) \qquad \beta_2 = A_{xy} - \frac{A_x A_y}{A_s}$$

$$\alpha_2 = B_{yz} - \frac{B_y B_z}{B_s} \qquad \beta_3 = A_y^2 + C_z^2 - \left(\frac{A_y^2}{A_s} + \frac{C_z^2}{C_s}\right)$$

$$\beta_1 = C_{zx} - \frac{C_x C_z}{C_s} \qquad \lambda_2 = A_x^2 + B_z^2 - \left(\frac{A_x^2}{A_s} + \frac{B_z^2}{B_s}\right)$$

When the above "A", "B" and "C" summation coefficients have been determined by calculation tabulation and summation of the element coefficients the combination coefficients of formulae (12a),(13a),(14a) can be found and set-up in conjunction with the three expansions already known. These three formulae then can be solved simultaneously for the only unknowns, the three end forces P,Q,N.

Next the end moments can be found by proper substitution of forces P,Q,N and the summation coefficients in formulae (6a), (7a),(8a) to determine the origin moments M_{Ox}, M_{Oy}, M_{Oz} and by further substitution of forces P,Q,N and of the values "L", "H", "T" (Figures 2 and 3) for "x", "y", "z" in formulae (26),(27) (28) to determine the far-end moments M_{Ax}, M_{Ay}, M_{Az}.

As described under the preceding single-plane instructions, each position of a simple element is shown by the Meyer-Hovgaard method on an individual form, together with the corresponding formulae for its "A" (or "B" or "C") coefficients, an identifying and orienting sketch, and a table of its principal dimensions and factors. All forms are numbered for convenient identification. Two additional forms permit the unhesitating selection of the corresponding forms for the two planes normal to the plane with which the element is associated. Another form permits the convenient tabulation of these form numbers opposite their appropriate elements, together with element length or bend radius and with element distance from each axis.

For determining the end forces and moments, there is a form for summation of the "A", "B", "C" coefficients taken from the element forms and for the tabulation of the resultant summation coefficients, a form for computation of the combination coefficients of formulae (12a) to (14a), forms for the computation and checking of the values for the end forces and moments, and a form on which these values may be summarized in conjunction with stress summaries for purposes of comparison and acceptability.

For determining the end stresses σ_{Ob}, σ_O, σ_{Od}, τ_O, σ_{Ab}, σ_A, σ_{Ad}, τ_A, and the resultant "equivalent stress" σ_{eq} in accordance with formulae (15) to (24) and (26) to (28) inclusive,* a form is provided for their tabulation after being run through a computing

* Note that formulae (18) and (22) may be neglected ordinarily.

machine. Thereafter they may be recorded on the summary form described in the preceding paragraph.

Figure 3 shows the same typical section of pipe line in space as Crocker and McCutchan analyze (I, page 629, Figure 23). However, where they take the origin "O" at the corner of the second bent section, the admonitions hereinafter given are obeyed in Figure 3 by taking the origin "O" at the lower end whose anchor originally is represented by reacting forces and resisting moments.

In this example, L = 11′ − 0″, H = 21′ − 6″, T = 9′ − 0″, the pipe size is 12″, the wall thickness t = ½″, the temperature rise is from 60° to 725°F, the radius of the bent elements # 2 and 4 is only 5 times the size or 60″, the lengths of the three straight elements # 1, 3, and 5 are 48″, 138″, and 72″, respectively. On this basis Hovgaard's flexibility factor K = 2.0, E is taken as 25,000,000 p.s.i. at 725°F, and the rectangular moment of inertia I is 361.5″⁴. The over-all thermal expansions are Δy = 0.530″, Δx = 1.270″, Δz = 0.647″.

Using five "Standard Forms" for each reference plane, the resultant summation coefficients are found to be:

Coefficients:	A_S	A_x	A_y	A_x^2	A_y^2	A_{xy}
"A"-plane (XY)	576.48	71837	54117	15,065,871	5,466,827	7,727,754
"B"-plane (YZ)	516.12	49527	9097	5,105,278	774,144	982,498
"C"-plane (ZX)	569.28	11023	83090	848,016	17,968,618	2,813,314

The combination coefficients from these are found to be:

α_1	α_2	β_1	β_2	β_3	λ_2
6,193,764	109,548	1,204,438	984,063	1,021,176	6,727,838

The resulting end-forces and end-moments, together with the values found by Crocker and McCutchan, are:

	P	Q	N	M_{Oz}	M_{Ox}	M_{Oy}
M—H	20,962#	3861#	5089#	−1,486,674″#	420,289″#	−336,883″#
C—McC	20,870	3855	5010	1,480,000	412,800	324,000
	F_y	F_z	F_x	M_{zy}	M_{xz}	M_{xy}

The close concordance is striking. It may be noted that in each case the Meyer-Hovgaard values are slightly higher and therefore tend toward the side of safety.

Determination of Maximum, Minimum, and Intermediate Conditions (for Location of Take-Down Joints)—The maximum conditions of stress are apt to occur at the fixed ends, just as for a beam with fixed ends.

However, if for instance the section of piping in Figure 1 is very long, it may require a flanged joint somewhere near mid-length either in order to get readily-procurable lengths of pipe or tubing or for convenience in handling during erection and maintenance. Visual inspection will show that the two ends of the bent element are points farthest away from a straight line joining the two fixed ends, and therefore are very liable to be points of high moment and stress. Obviously, the optimum location for any intermediate joint is at a point of zero moment; and conversely, the location to avoid, if at all possible, is any point of high moment.

Similarly, if the section of piping in Figure 2 is very long or awkward to handle and an intermediate joint is required, visual inspection of the various elements with respect to a straight line joining the two fixed ends will show that the middle bent element is farthest away from the line and therefore its end points are very liable to reveal high moments and stresses. This is most clearly indicated by inspection of projections "B" and "C" of Figure 2 where the middle bent element is revealed as farther from the line than either of the other bent elements.

Single plane sections of piping like Figure 1 can be tested easily for points of zero moment as follows: At such points formula (25) $M_s = M_o + Py - Qx$ is equal to zero, and x and y are the only unknowns. Transposing and dividing by P, there results $y = \frac{Q}{P} x - \frac{M_o}{P}$. This is an equation of a straight line in the XY plane, having a slope $\frac{dy}{dx} = \frac{Q}{P}$ and an intercept on the Y-axis of $-\frac{M_o}{P}$. Sketching this line in on the XY plane will determine any points of inter-section with the line of the piping section.

Three-plane sections of piping like Figure 2 can be tested just as readily for points of zero moment by the above method in the following manner. The bending moment at any point "S" is found by writing formula (15): $M_S = \sqrt{M^2_{Sx} + M^2_{Sz}}$. The moments under the radical sign are chosen from formulae (26), (27), (28) according to the axes of the plane normal to the piping section at the point "S". However $M_S = 0$. Since only rational values and conditions exist in piping calculations, the only condition under which $M_S = 0$ is when each moment under the radical equals zero. Therefore the two applicable moment formulae from (26), (27), (28) are equated each to zero, transposed into line equations duly sketched in on the proper projections "A", "B", or "C" of Figure 2, and their possible intersections with the line of the piping section determined.

Taking briefly an example from Figure 2 and assuming that there is a possible point of zero moment somewhere in the second straight element, reference to projection "A" of Figure 2 shows this element to be parallel to the X axis. The axes of the plane normal to this element are the Y and Z axes. Writing formulae (26), (28) and (15) accordingly:

$$M_{Sz} = M_{Oz} + Py - Qx \qquad (26')$$

$$M_{Sy} = M_{Oy} + Nx - Pz \qquad (28')$$

$$M_S = \sqrt{M^2_{Sz} + M^2_{Sy}} = 0 \therefore M_{Sz} = M_{Sy} = 0 \qquad (15')$$

The "O"-end moments and the end forces are known. Therefore the line equations become:

$$y = \frac{Q}{P}x - \frac{M_{Oz}}{P}, \text{ a line in the XY plane.}$$

$$x = \frac{P}{N}z - \frac{M_{Oy}}{N}, \text{ a line in the ZX plane.}$$

It is evident that the second line when shown on the "C" projection, Figure 2, will not intersect the element. Also, it is evident in this example that the element in question lies in the XY plane at a constant and known distance from the X-axis, i.e. $z = 0$ and $y =$ a constant. Therefore, formulae (26') and (28') can be solved without the necessity of further transposition. The

location thus established is an optimum one for a flanged joint since it is taken at a point where M_S is minimum since $M_{Sz} = 0$ and M_S becomes M_{Sy} which is at a minimum since $z = 0$.

Application.

Instructions for Use of the Standard Forms—There are three distinct groups of the Meyer-Hovgaard method "Standard Forms". The first group is applicable to both single-plane and three-plane piping. It consists entirely of forms for the elements. The second group is applicable to three-plane sections of piping only. The third group is applicable to single-plane sections of piping only.

The application of these forms to the solution of thermal stresses in piping already has been described in general terms. The following instructions cover the detail procedure and application of the "Standard Forms" and the Meyer-Hovgaard method. The "Standard Forms" are numbered for ease of identification.

For convenience in making up sets of forms in complete and orderly fashion and in the order of their use, the following tabulation by groups and titles is given. A copy of each "Standard Form" is included at the end of this article.

Title	Group: Single-Plane	Both	Three-Plane
Instruction Form:			
Gen. Arrgt. of Planes	—	—	47
Common Forms:			
Dimensions	53	—	51
Tab. of Form Nos.&Dims.	50(except Y-Z and Z-X columns)*		50*
Coefficients of Section		1 to 16, incl.	17 to 45, incl.
Coefficients of Section	46(Plane XY only)	—	46
Coefficients of Section	—	—	48
Determination of Forces,	100	—	102, 103A, 104
and Moments	101	—	105, 106
Forces, Moments and Stresses	54, 112	—	52
*Reference Forms:			
Chart for Selection of Forms	55		55
Table for Selection of Forms	—		49

The procedure for solving a single-plane section of piping is first to sketch the section on Form No. 53 so that one end is at "O" if possible, or otherwise on either the X-axis or Y-axis. These axes can be taken at the convenience of the investigator, without regard to the relative location of the piping section on the plans or in the plant. It will be found convenient to keep the line of the piping from crossing either axis. The sketch next is split up into consecutive elements, either straight or bent, which are numbered beginning at the "O" end. To the sketch are added the distances of that end of each element nearer the origin "O" from each axis. These distances are denoted as "L_x" and "L_y" on Form No. 50. Finally the table of dimensions on Form No. 53 should be worked up and filled in.

As indicated above, Form No. 50 next is to be filled in, using only the six left-hand columns, listing the elements consecutively by their numbers. The element length "L" or bend radius "R" and the axis distances "L_x" and "L_y" are available from the sketch of Form No. 53. The "form number" of the correct "Standard Form" for each element may be determined by the aid of Reference Form No. 55, and listed.

The third step is to select a copy of the "Standard Form" which matches each element, fill in the corresponding dimensions from Form No. 50, and complete the calculations of the element coefficients A_s, A_x, A_y, A_x^2, A_y^2, A_{xy}.

These element coefficients next are tabulated and added up on Form No. 46. Since only one plane is involved, the lines marked "Plane YZ" and "Plane ZX" are crossed-out.

The summaries on Form No. 46 provide the summation coefficients necessary to fill out the basic formulae at the top of form No. 100. Solution of the formulae as provided on the Form gives the values for the end forces P and Q. These in turn are checked at the top of Form No. 101 and if satisfactorily close are used for determining the end moment M, which in turn is checked.

The summation coefficients from Form No. 46 and the end forces and moments from Forms Nos. 100 and 101 are entered on Form No. 54 and used to determine the stresses tabulated thereon for each end of the section of piping.

If other points of the section of piping are to be investigated, the table and space at the lower part of Form No. 112 may be used.

Forms Nos. 1 to 16 cover: (a) straight elements which are parallel to or at 45° to one of the reference axes; and (b) bent elements of 90° in four different positions and of 45° in eight different positions relative to the reference axes. Where the element actually deviates by a few degrees from the above positions and angle of bend, no appreciable error is introduced by using Forms Nos. 1 to 16. Where the deviation is greater than a few degrees, "Special Forms" can be made up. Where the use of some special angle of bend becomes usual or standard practice, the corresponding "Special Forms" should be added to the "Standard Forms".

The procedure for solving a three-plane section of piping follows the procedure for a single-plane section exactly, except that three planes involves three sets of forms for the element coefficients, three summary sheets (one for each set), and an additional form for the combination coefficients. Form No. 47 will be found useful in setting up the three reference planes in connection with the given section of piping. The admonitions already given for a single-plane section apply here with greater emphasis:

(a) place one end at point "O" if possible, otherwise place it on the corresponding reference axis;

(b) take the reference axes and planes without regard to the actual position which the section is to occupy on the job; the axes and planes should be taken as most convenient to the investigator;

(c) keep the line of the piping section within the "box", i.e. see the line does not cross any reference axis.

In determining the best combination of section of piping and reference planes, the "A", "B", "C" projections are sketched in on the respective XY, YZ, ZX planes of Form No. 51, the successive elements shown and numbered beginning at the "O" end, their lengths "L" or bend radius "R" indicated, and the axis distances "L_x", "L_y", "L_z" corresponding to axes of the projection shown. The table of dimensions likewise is worked up and filled in.

Using the three sketches on Form No. 51 and the list and chart of forms on reference Forms Nos. 49 and 55, all columns of Form No. 50 are filled in. The elements are put down in numerical order and the dimensions and form numbers back-checked before proceeding further.

The next step is to make up three sets of copies of "Standard Forms",—a set per plane complete as listed on Form No. 50 and equal in number to the number of elements. On the "Standard Forms" are entered the corresponding plane and element number, the applicable dimensions from Forms Nos. 50 and 51, the resultant factors tabulated on each "Standard Form", and finally the element coefficients computed from the above assembled data.

When these sets of forms are completed, a summary of all element coefficients for each set (and plane) is made on copies of Form No. 46, the summation coefficients determined and entered, and a check mark placed opposite the proper plane.

Using the "A", "B", "C" summation coefficients from the three Forms No. 46 just filled out, the combination coefficients are worked out on Form No. 48.

These combination coefficients next are used to set up the three simultaneous end-force equations at the top of Form No. 102. A method of solution is provided on Forms Nos. 102 and 103A. A check-up method is provided on Form No. 104. These forms furnish a convenient record of solution. This method of solution and of check-up need not be followed, however, if the investigator prefers some other.

Having determined the end-forces P,Q,N, the values of the end-moments can be found by the method provided on Form No. 105 and checked on Form No. 106.

There remains only the computation and tabulation of the unit stresses. Form No. 113 provides for the detail computation of these stresses for both ends of the section of piping, employing cross-section data from Form No. 51, end-forces from Form No. 103-A, and end-moments from Form No. 105. Care should be taken to indicate which end-moments are combined for bending and which end-moment is used for torsion, and these checked against the projection sketches of Form No. 51. Finally the fiber stresses thus determined are entered on summary Form No. 52, together with the combination coefficients used to solve for forces P,Q,N, the primary summation coefficients used to solve for end-moments, the final values for forces P,Q,N, and the final values for the end-moments.

If it is desirable to determine the conditions at some intermediate point S(x,y,z) of the section of piping, the point-moments

can be found by the moment formulae given on instruction Form No. 47. The end-forces being external stresses are the same from one end to the other of the section of piping.

Sometimes there is more than one elevated-temperature operating condition for a section of piping, as for instance the branch to a unit intended to run with either superheated or saturated steam, depending upon which is available. Again a change in design may include a change in total-temperature conditions. A convenient Form No. 114 has been included to enable the determination of the effect, on the piping thermal stresses, of such changes.

De Jonge and Smith Basic Formulae—Mr. A. E. R. de Jonge submitted in his discussion on an A.S.M.E. paper (III) basic formulae for the values of rotation and deflection of smooth pipe bends subtended by any angle α and acted upon by either forces or couples in the plane of the bend or in one of the two reference planes normal to the plane of the bend. As Appendix No. 3 of a subsequent paper (IV) Mr. S. Smith developed these formulae into twelve separate cases and tabulated corresponding moment, rotation, and deflection values for angles of 22½°, 30°, 45°, 60°.

Furthermore, finding nowhere similar formulae for straight elements adjacent to the above bent elements and therefore at any angle θ with the horizontal axis, corresponding to the angle α of the bent element, Mr. S. Smith developed as a corollary Appendix No. 4 three cases covering moments and forces in one of the two reference planes normal to the plane of the straight element.

These formulae are very useful in preparing "Special Forms" and are reproduced here for convenience and ready reference. The Hovgaard flexibility factor "K"(4) has been substituted for the reciprocal Walker-Crocker rigidity multiplication factor.

The formulae for bent elements of any angle α are as follows, expressed in terms of an element in the ZX-plane:

Case I: Force F parallel to X-axis, in direction of Z-axis. Bent element fixed at Z-axis end.

Moment $= FR\ (1 - \cos \alpha)$.

$$\Delta x = \frac{KFR^3}{EI} [\alpha(0.5 + \cos^2\alpha) - 1.5 \sin \alpha \cos \alpha]$$

$$\psi_t = \frac{KFR^2}{EI} (\sin \alpha - \alpha \cos \alpha).$$

$$\Delta z = \frac{KFR^3}{EI} (0.5\sin^2\alpha - \alpha \sin \alpha \cos \alpha + \cos \alpha - \cos^2\alpha).$$

Angle	Fixed End	Moment at ψ_t =	Rotation	Δx—Deflection	Δz—Deflection
60°	0.500	FR	0.342	KFR^2/EI	0.397R ψ_t	0.501R ψ_t
45°	0.293	"	0.152	"	0.233R "	0.424R "
30°	0.134	"	0.047	"	0.106R "	0.304R "
22½°	0.076	"	0.200	"	0.045R "	0.260R "

Case II: Force F in X-axis, in direction of origin.
Bent element fixed in plane at other end, α from X-axis.

$$\text{Moment} = FR \sin \alpha. \quad \Delta x = \frac{KFR^3}{EI} (0.5\alpha - 0.5 \sin \alpha \cos \alpha).$$

$$\psi_t = \frac{KFR^2}{EI} (1 - \cos \alpha). \quad \Delta z = \frac{KFR^3}{EI} (1 - \cos \alpha - 0.5 \sin^2\alpha).$$

Angle	Fixed End	Moment at ψ_t =	Rotation	Δx—Deflection	Δz—Deflection
60°	0.866	FR	0.500	KFR^2/EI	0.614R ψ_t	0.250R ψ_t
45°	0.707	"	0.293	"	0.487R "	0.147R "
30°	0.500	"	0.134	"	0.338R "	0.067R "
22½°	0.383	"	0.077	"	0.254R "	0.040R "

Case III: Force F parallel to Z-axis, in direction away from X-axis.
Bent element fixed at Z-axis end.

$$\text{Moment} = FR \sin \alpha. \quad \Delta x = \frac{KFR^3}{EI}$$

$$(1.5 \sin^2\alpha - \alpha \sin \alpha \cos \alpha + \cos \alpha - 1)$$

$$\psi_t = \frac{KFR^2}{EI} (\alpha \sin \alpha + \cos \alpha - 1). \quad \Delta z = \frac{KFR^3}{EI}$$

$$[\alpha(0.5 + \sin^2\alpha) + 1.5 \sin \alpha \cos \alpha - 2 \sin \alpha]$$

Angle	Moment at Fixed End	ψ_t = Rotation		Δx—Deflection	Δz—Deflection
60°	0.866 FR	0.407	KFR^2/EI	0.364R ψ_t	0.555R ψ_t
45°	0.707 "	0.262	"	0.244R "	0.463R "
30°	0.500 "	0.128	"	0.117R "	0.330R "
22½°	0.383 "	0.074	"	0.065R "	0.253R "

Case IV: Force F normal to X-axis, in direction away from X-axis.

Bent element fixed in plane at other end, α from X-axis.

Moment = FR $(1 - \cos \alpha)$.

$$\Delta_x = \frac{KFR^3}{EI} (1 - \cos \alpha - 0.5 \sin^2 \alpha).$$

$$\psi_t = \frac{KFR^2}{EI} (\alpha - \sin \alpha). \quad \Delta_z = \frac{KFR^3}{EI}$$

$$(1.5\alpha - 2 \sin \alpha + 0.5 \sin \alpha \cos \alpha).$$

Angle	Moment at Fixed End	ψ_t = Rotation		Δx—Deflection	Δz—Deflection
60°	0.500 FR	0.181	KFR^2/EI	0.691R ψ_t	0.306R ψ_t
45°	0.293 "	0.078	"	0.551R "	0.179R "
30°	0.134 "	0.024	"	0.375R "	0.079R "
22½°	0.077 "	0.010	"	0.300R "	0.057R "

Case V: Couple M in ZX-plane at free end.

Bent element fixed at Z-axis end.

Moment = a constant, M. $\Delta x = \frac{KMR^2}{EI} (\sin \alpha - \alpha \cos \alpha)$.

$$\psi_m = \alpha \frac{KMR}{EI} \quad \Delta z = \frac{KMR^2}{EI} (\alpha \sin \alpha + \cos \alpha - 1).$$

Angle	ψ_m = Rotation		Δx—Deflection	Δz—Deflection
60°	1.047	KMR/EI	0.327R ψ_m	0.389R ψ_m
45°	0.785	"	0.193R "	0.334R "
30°	0.524	"	0.089R "	0.244R "
22½°	0.393	"	0.051R "	0.188R "

Case VI: Couple M in ZX-plane at free end at X-axis.
Bent element fixed in plane at other end, α from X-axis.

Moment = a constant, M. $\quad \Delta x = \frac{KMR^2}{EI}(1 - \cos \alpha).$

$$\psi_m = \alpha \frac{KMR}{EI}. \quad \Delta z = \frac{KMR^2}{EI}(\alpha - \sin \alpha).$$

Angle	ψ_m = Rotation	Δx—Deflection	Δz—Deflection
60°	1.047 KMR/EI	0.477R ψ_m	0.173R ψ_m
45°	0.785 "	0.373R "	0.100R "
30°	0.524 "	0.256R "	0.045R "
22½°	0.393 "	0.194R "	0.026R "

Case VII: Force F normal to ZX-plane at free end in direction of Y-axis.
Bent element fixed at Z-axis end.

$$\mu = 0.3 \qquad \psi_{Fx} = \frac{FR^2}{EI}[(1 + 0.5\,\mu)(\sin \alpha - \alpha \cos \alpha)]$$

$$M_z = FR \sin \alpha$$

$$\psi_{Fz} = \frac{FR^2}{EI}[(1 + 0.5\,\mu)\ \alpha\ \sin \alpha - (1 + \mu)(1 - \cos \alpha)]$$

$$M_x = FR(1 - \cos \alpha). \quad \Delta_y = \frac{FR^3}{EI}[(2 + 1.5\mu)\ \alpha + 0.5 \sin \alpha \cos \alpha - 2(1 + \mu) \sin \alpha]$$

Angle	M_z	M_x	ψ_{Fz} — Rotation	ψ_{Fx} — Rotation	Δ_{Yz} — Deflection	Δ_{Yx} — Deflection
60°	0.866 FR	0.500 FR	0.393 FR^2/EI	0.394 FR^2/EI	0.481R ψ_{Fz}	0.481R ψ_{Fx}
45°	0.707 "	0.293 "	0.258 "	0.175 "	0.372R "	0.372R "
30°	0.500 "	0.134 "	0.127 "	0.054 "	0.264R "	0.264R "
22½°	0.383 "	0.076 "	0.074 "	0.023 "	0.210R "	0.210R "

Case VIII: Force F normal to ZX-plane at free end in direction of Y-axis.
Bent element free end at X-axis, fixed end in plane.

$M_z = FR \sin \alpha.$ $\qquad \psi_{Fz} = \frac{FR^2}{EI}[(1 + 0.5\,\mu)\ \alpha - (1 + \mu) \sin \alpha + 0.5\,\mu \sin \alpha \cos \alpha]$

$M_x = FR\ (1 - \cos \alpha)$.

$$\psi_{Fx} = \frac{FR^2}{EI}\ [(1 + \mu)\ (1 - \cos \alpha) - 0.5\ \mu \sin^2 \alpha]$$

$$\Delta_y = \frac{FR^3}{EI}\ [(2 + 1.5\ \mu)\ \alpha + 0.5 \sin \alpha \cos \alpha - 2\ (1 + \mu) \sin \alpha]$$

Angle	M_z	M_x	ψ_{Fz} — Rotation	ψ_{Fx} — Rotation	Δ_{Yz} — Deflection	Δ_{Yx} — Deflection
60°	0.866 FR	0.500 FR	0.143 FR^2/EI	0.537 FR^2/EI	0.557R ψ_{Fz}	0.557R ψ_{Fx}
45°	0.707 "	0.293 "	0.056 "	0.306 "	0.442R "	0.442R "
30°	0.500 "	0.134 "	0.017 "	0.137 "	0.310R "	0.310R "
22½°	0.383 "	0.077 "	0.009 "	0.077 "	0.238R "	0.238R "

Case IX: Couple M in plane normal to ZX-plane and parallel to YZ-plane at free end.

Bent element fixed at Z-axis end.

Moment = a constant, M. $\psi_{Mx} = \frac{MR}{EI}$

$$[(1 + 0.5\ \mu)\ \alpha + 0.5\ \mu \sin \alpha \cos \alpha]$$

$$\psi_{Mz} = -\frac{MR}{2EI}\mu \sin^2 \alpha.$$

$$\Delta_y = \frac{MR^2}{EI}[(1 + 0.5\ \mu)\ (\sin \alpha - \alpha \cos \alpha)]$$

Angle	ψ_{Mz} — Rotation	ψ_{Mx} — Rotation	Δ_{Mz} — Deflection	Δ_{Mx} — Deflection
60°	−0.113 MR/EI	1.269 MR/EI	0.341R ψ_{Mz}	0.341R ψ_{Mx}
45°	−0.075 "	0.978 "	0.193R "	0.193R "
30°	−0.038 "	0.667 "	0.085R "	0.085R "
22½°	−0.022 "	0.505 "	0.047R "	0.047R "

Case X: Couple M in plane normal to ZX-plane and parellel to YZ-plane at free end.

Bent element free end on X-axis.

Moment = a constant, M. $\psi_{Mx} = +\frac{MR}{EI}$

$$[(1 + 0.5\ \mu)\ \alpha - 0.5\ \mu \sin \alpha \cos \alpha]$$

$$\psi_{Mz} = -\frac{MR}{2EI}\,\mu \sin^2 \alpha.$$

$$\Delta_y = \frac{MR^2}{EI}\,[(1 + \mu)\,(1 - \cos \alpha) - 0.5\,\mu \sin^2 \alpha]$$

Angle	ψ_{Mz} — Rotation		ψ_{Mx} — Rotation		Δ_{Mz} — Deflection		Δ_{Mx} — Deflection	
60°	−0.113	MR/EI	1.239	MR/EI	0.477R	ψ_{Mz}	0.477R	ψ_{Mx}
45°	−0.075	"	0.828	"	0.406R	"	0.406R	"
30°	−0.038	"	0.537	"	0.247R	"	0.247R	"
22½°	−0.022	"	0.399	"	0.204R	"	0.204R	"

Case XI: Couple M in plane normal to ZX-plane and parallel to XY-plane at free end.
Bent element fixed at Z-axis end.

Moment = a constant, M. $\psi_{Mz} = +\frac{MR}{EI}$

$$[(1 + 0.5\,\mu)\,\alpha - 0.5\,\mu \sin \alpha \cos \alpha]$$

$$\psi_{Mx} = -\frac{MR}{2EI}\,\mu \sin^2 \alpha.$$

$$\Delta_y = \frac{MR^2}{EI}[(1 + 0.5\,\mu)\,\alpha \sin \alpha - (1 + \mu)\,(1 - \cos \alpha)]$$

Angle	ψ_{Mz} — Rotation		ψ_{Mx} — Rotation		Δ_{Mz} — Deflection		Δ_{Mx} — Deflection	
60°	1.139	MR/EI	−0.113	MR/EI	0.383R	ψ_{Mz}	0.383R	ψ_{Mx}
45°	0.828	"	−0.075	"	0.342R	"	0.342R	"
30°	0.537	"	−0.038	"	0.254R	"	0.254R	"
22½°	0.398	"	−0.022	"	0.196R	"	0.196R	"

Case XII: Couple M in XY-plane at free end.
Bent element free end on X-axis.

Moment = a constant, M. $\psi_{Mz} = +\frac{MR}{EI}$

$$[(1 + 0.5\,\mu)\,\alpha + 0.5\,\mu \sin \alpha \cos \alpha]$$

$$\psi_{Mx} = -\frac{MR}{2EI}\mu \sin^2 \alpha. \qquad \Delta_y = \frac{MR^2}{EI}\,[(1 + 0.5\,\mu)$$

$$(\alpha - \sin \alpha) - 0.5\,\mu \sin \alpha\,(1 - \cos \alpha)]$$

Angle	ψ_{Mz} — Rotation	ψ_{Mx} — Rotation	Δ_{Mz} — Deflection	Δ_{Mx} — Deflection
60°	1.269 MR/EI	−0.113 MR/EI	0.124R ψ_{Mz}	0.124R ψ_{Mx}
45°	0.978 "	−0.075 "	0.065R "	0.065R "
30°	0.667 "	−0.037 "	0.027R "	0.027R "
22½°	0.505 "	−0.027 "	0.015R "	0.015R "

The formulae for straight elements of length L included at any angle θ to a horizontal plane and having the fixed end at the origin "O", expressed in terms of an element in the XY-plane, are as follows:

Case A: Couple M in plane normal to XY-plane and parallel to YZ-plane. Moment = a constant, M_x.

$$\psi_x = \text{rotation along X-axis} = \frac{M_x L}{EI}(1.15 + 0.15 \cos 2\theta)$$

$$\Delta_z = \frac{M_x L^2}{2EI} \sin\theta = \psi_x \frac{2.3 + 0.3 \cos 2\theta}{L \sin\theta}$$

Case B: Couple M in plane normal to XY-plane and parallel to ZX-plane. Moment = a constant, M_y.

$$\psi_y = \text{rotation along Y-axis} = \frac{M_y L}{EI}(1.15 - 0.15 \cos 2\theta)$$

$$\Delta_z = \frac{M_y L}{EI}(1.15 - 0.15 \cos 2\theta) = \psi_y \frac{L \cos\theta}{2.3 - 0.3 \cos 2\theta}$$

Case C: Force F normal to ZX-plane containing the element, and in direction of Y-axis.

$$\psi_z = \text{rotation about Z-axis} = \frac{FL^2}{2EI}\cos\theta\,(1.15 - 0.15\cos 2\theta).$$

$$\psi_x = \text{rotation about X-axis} = \frac{FL^2}{2EI}\sin\theta\,(1.15 + 0.15 \cos 2\theta).$$

$$\Delta_y = \frac{FL^3}{3EI}$$

Tables of Section Factors—For the convenience of all who have thermal expansion stress calculations to make for piping, Tables III, IV and V have been worked up.

Tubing is used in Naval piping systems, rather than pipe. Table III lists cross-sectional dimensions and Hovgaard flexibility data for 635 p.s.i. 850°F molybdenum alloy steel tubing in accordance with Naval specifications. A bend radius of six times the size (commonly stated as "six diameters") is the minimum permitted by Naval piping specifications.

Table IV lists cross-sectional dimensions and Hovgaard flexibility data for 400 p.s.i. This may be either plain steel tubing for temperatures up to 650°F or molybdenum alloy steel tubing for temperatures up to 850°F, in either case in accordance with Naval specifications.

Table V lists cross-sectional dimensions and Hovgaard flexibility data for steel pipe, either plain or alloy, in accordance with the American Standards Association Standard B36.10. This Standard lists "nominal" (i.e., average) wall thicknesses for the various "schedule numbers" which include an allowance for a mill tolerance of 12.5 per cent under the "nominal" thickness. Since Naval design is based on the minimum dimension as representing the worst possible condition that actually will be encountered and since it is common sense and prudence to apply the same viewpoint to power plant piping, Table V is based on minimum wall thicknesses, and corresponding maximum inside diameters, only 87.5 per cent of the "nominal" wall thicknesses.

In this connection, (IV) is quoted in part:

"In calculating the deflections, the value for I was determined by the use of the measured thicknesses of the walls of the quarter-bend and of the tangents. * * It was found that the use of correct values for wall thickness made a difference of about 10 per cent as compared with the results obtained by the use of nominal thicknesses."

Since other assumptions may make nearly as much difference as that quoted, it seems inadvisable to include a further known discrepancy which may be avoided by using minimum dimensions rather than average figures. Therefore, it is believed that all similar tables in existence elsewhere are in error in this respect and should not be used for thermal expansion stress determination unless suitable modification is made for minimum dimensions.

TABLE III 600 p.s.i. 850° F C-Mo Steel
Navy Dept. Specs. 44T33

Size	d_o	d_i	A	r_o	r_t^*	t^*	h	r	I	R_6	λ_6	K_6
Size	Inches	Inches	Sq. In.	Inches	Inches	Inches	Inches	Inches	$(\text{Inches})^4$	Inches		
2 1/2	2.875	2.551	1.381	1.438	1.276	.162	.081	1.357	1.275	.15	1.320	1.411
3	3.500	3.106	2.044	1.750	1.553	.197	.099	1.652	2.798	18	1.299	1.424
3 1/2	4.000	3.500	2.668	2.000	1.775	.225	.113	1.888	4.771	21	1.325	1.408
4	4.500	3.994	3.375	2.250	1.997	.253	.127	2.124	7.640	24	1.346	1.396
4 1/2	5.000	4.436	4.180	2.500	2.218	.282	.141	2.359	11.6715	27	1.368	1.384
5	5.563	4.937	5.158	2.782	2.469	.313	.157	2.626	17.855	30	1.326	1.387
5 1/2	6.125	5.435	6.265	3.063	2.718	.345	.173	2.891	26.26	33	1.362	1.387
6	6.625	5.879	7.300	3.313	2.940	.373	.187	3.127	35.93	36	1.373	1.381
7	7.625	6.767	9.700	3.813	3.364	.429	.215	3.599	63.00	42	1.391	1.372
8	8.625	7.653	12.425	4.313	3.827	.486	.243	4.070	103.265	48	1.499	1.322
9	9.625	8.451	16.665	4.813	4.226	.542	.271	4.497	170.9	54	1.447	1.344
10	10.750	9.540	19.28	5.375	4.770	.605	.303	5.073	248.95	60	1.410	1.362
11	11.750	10.426	23.06	5.875	5.213	.662	.331	5.544	355.65	66	1.422	1.356
12	12.750	11.314	27.14	6.375	5.657	.718	.359	6.016	492.9	72	1.428	1.353

*Based on the minimum wall thickness t and the resultant maximum inside diameter

TABLE IV 400 p.s.i. 650°F Steel (650°F C-Mo)
Navy Dept. Specs. 44T13

Size	d_o	d_i	A	r_o	r_i^*	t^*	h	r	I	R_6	λ_6	K_6
	Inches	Inches	Sq. In.	Inches	Inches	Inches	Inches	Inches	$(\text{Inches})^4$	Inches		
2 1/2	2.875	2.635	1.039	1.438	1.318	.120	.060	1.378	0.987	15	0.948	1.767
3	3.500	3.210	1.528	1.750	1.605	.145	.073	1.678	2.1545	18	0.927	1.796
3 1/2	4.000	3.670	1.988	2.000	1.835	.165	.083	1.918	3.6615	21	0.942	1.773
4	4.5000	4.128	2.520	2.250	2.064	.186	.093	2.157	5.875	24	0.959	1.748
4 1/2	5.000	4.588	3.103	2.500	2.294	.206	.103	2.397	8.9295	27	0.968	1.735
5	5.563	5.103	3.849	2.782	2.552	.230	.115	2.667	13.73	30	0.970	1.732
5 1/2	6.125	5.619	4.667	3.063	2.810	.253	.127	2.937	20.155	33	0.968	1.735
6	6.625	6.079	5.448	3.313	3.040	.273	.137	3.177	27.525	36	0.974	1.727
7	7.625	6.995	7.234	3.813	3.498	.315	.158	3.656	48.41	42	0.990	1.705
8	8.625	7.913	9.248	4.313	3.957	.356	.178	4.135	79.19	48	0.999	1.694
9	9.625	8.831	11.510	4.813	4.416	.397	.199	4.615	122.75	54	1.007	1.683
10	10.750	9.862	14.375	5.375	4.931	.444	.222	5.153	191.2	60	1.003	1.688
11	11.750	10.780	17.165	5.875	5.390	.485	.243	5.633	272.75	66	1.009	1.681
12	12.750	11.698	20.20	6.375	5.849	.526	.263	6.112	378.0	72	1.014	1.675
14	14.000	12.844	24.37	7.000	6.422	.578	.289	6.711	549.85	84	1.078	1.602
15	15.000	13.762	27.96	7.500	6.881	.619	.310	7.191	724.3	90	1.077	1.603
16	16.000	14.678	31.855	8.000	7.339	.661	.331	7.670	938.55	96	1.079	1.601

*based on the minimum wall thickness t and the resultant maximum inside diameter

TABLE V A.S.A. Standard B36.10

Size	d_o	d_i	A	r_o	r_t^*	t^*	h	r	I	R_6	λ_6	K_6
	Inches	Inches	Sq. In.	Inches	Inches	Inches	Inches	Inches	$(\text{Inches})^4$	Inches		
Schedule 40:												
2 1/2	2.875	2.520	1.504	1.438	1.260	.178	.089	1.349	0.88345	15	1.467	1.336
3	3.500	3.122	1.966	1.750	1.561	.189	.095	1.656	2.7035	18	1.241	1.462
3 1/2	4.000	3.604	2.355	2.000	1.802	.198	.099	1.901	4.786	21	1.151	1.533
4	4.500	4.086	2.791	2.250	2.043	.207	.104	2.147	6.448	24	1.078	1.606
5	5.563	5.108	3.809	2.782	2.554	.228	.114	2.668	13.598	30	0.961	1.745
Schedule 60:												
8	8.625	7.916	9.210	4.313	3.958	.355	.178	4.136	78.919	48	0.996	1.697
10	10.750	9.874	14.190	5.375	4.937	.438	.219	5.156	189.00	60	0.989	1.707
Schedule 80:												
3	3.500	2.976	2.665	1.750	1.488	.262	.131	1.619	3.512	18	1.799	1.226
3 1/2	4.000	3.444	3.250	2.000	1.722	.278	.139	1.861	6.269	21	1.686	1.256
4	4.500	3.910	3.897	2.250	1.955	.295	.148	2.103	11.480	24	1.601	1.283
5	5.563	4.908	5.382	2.782	2.454	.328	.164	2.618	18.535	30	1.436	1.350
6	6.625	5.870	7.410	3.313	2.935	.378	.189	3.124	36.290	36	1.394	1.370
10	10.750	9.712	16.68	5.375	4.856	.519	.260	5.116	218.9	60	1.190	1.500
12	12.750	11.548	22.94	6.375	5.774	.601	.301	6.075	424.35	72	1.172	1.515
14	14.000	12.688	27.50	7.000	6.344	.656	.328	6.672	613.75	84	1.238	1.464
16	16.000	14.524	35.358	8.000	7.262	.738	.369	7.631	1032.95	96	1.217	1.479
Schedule 100:												
8	8.625	7.488	13.205	4.313	3.794	.519	.260	4.054	108.95	48	1.516	1.315
10	10.750	9.494	19.97	5.375	4.747	.628	.314	5.061	266.75	60	1.471	1.334
12	12.750	11.274	27.85	6.375	5.637	.738	.369	6.006	504.3	72	1.473	1.333

*based on 12.5 per cent tolerance under the nominal thickness.

Conclusion and Acknowledgment—The present article, in introducing the Meyer-Hovgaard method for the first time to the "piping fraternity" at large, purposely has been restricted to an elementary exposition of the method. Application to relatively simple sections of piping in one plane and in three planes has been the view-point. Application to branched sections of piping has been left as more suitably a separate subject for a sequel article.

The policy for Naval vessel high temperature steam piping at this time is to simplify matters at the mill and in design by restricting piping material to one basic thickness per size per basic pressure. Except for one aberration, the basic pressures agree with the American Standards Association and are 150, 300, 400, 635, 900, and 1500 p.s.i. The pure economics of the situation undoubtedly is doing and will do the same to merchant vessel and power plant piping thicknesses.

However, should occasion arise where a section of piping is to consist of more than one thickness of material per size, proper allowance can be made on the respective element forms for the difference in flexibility factor "K" and in the solution for forces, moments, and unit stresses for the difference in cross-sectional dimensions at the various points investigated.

Variation in bend radius is not considered as a complication and automatically disposes of itself as the "Standard Forms" are applied to the successive elements.

Naval design has frowned with good reason upon creased and corrugated elements of piping sections. Consequently there is no attempt nor intention to include these in the present data.

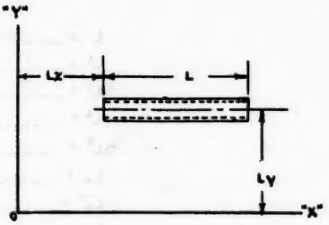
In conclusion the writer wishes to acknowledge gratefully the opportunity and inspiration given by Lieutenant Commander Guy Chadwick, U.S.N., the unselfish interest and cooperation of Mr. Rudolph Michel in reviewing this article at a time when many other demands were being made upon him, and the aid of associates in the preparation of the text, figures and tables.

In the preparation of this exposition, a copyrighted description (XI) of the method has been consulted liberally and is acknowledged appreciatively.

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L = _____
L^2 = _____
$\frac{1}{2}L^2$ = _____
L^3 = _____
$\frac{1}{3}L^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____

$A_s = L$ = _____

$A_x = LL_x + \frac{1}{2}L^2$

= _____ + _____ = _____

$A_y = LL_y$ = _____

$A_{x^2} = LL_x^2 + \frac{1}{3}L^3 + L^2 L_x$

= _____ + _____ + _____ = _____

$A_{y^2} = LL_y^2$ = _____

$A_{xy} = LL_x L_y + \frac{1}{2}L^2 L_y$

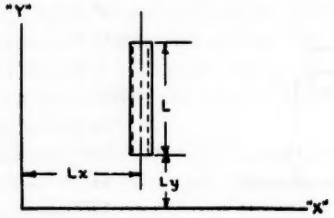
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 1 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L = _____
L^2 = _____
$\frac{1}{2}L^2$ = _____
L^3 = _____
$\frac{1}{3}L^3$ = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

As = L = _____ _____ _____ _____

Ax = LLx = _____ _____ _____ _____

Ay = $LLy + \frac{1}{2}L^2$

= _____ + _____ = _____ _____

Ax^2 = LLx^2 = _____ _____ _____ _____

Ay^2 = $LLy^2 + \frac{1}{3}L^3 + L^2Ly$

= _____ + _____ + _____ = _____ _____

Axy = $LLxLy + \frac{1}{2}L^2Lx$

= _____ + _____ = _____ _____

GROUP Nº ____ FORM Nº 2 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
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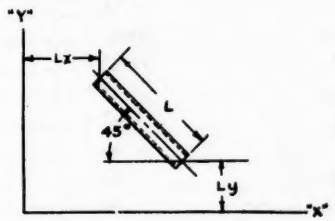
SECTION Nº ___ ___ PLANE



L = _____
L^2 = _____
$.354\,L^2$ = _____
L^3 = _____
$1/6\,L^3$ = _____
$1/12\,L^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____
$.707L^2$ = _____

As = L = _____

Ax = $LL_x + .354L^2$ = _____ + _____ = _____

Ay = $LL_y + 354L^2$ = _____ + _____ = _____

$Ax^2 = LL_x^2 + .707L^2L_x + 1/6L^3$

= _____ + _____ + _____ = _____

$Ay^2 = LL_y^2 + .707L^2L_y + 1/6L^3$

= _____ + _____ + _____ = _____

$Axy = LL_xL_y + .354L^2L_x + .354L^2L_y + 1/12\,L^3$

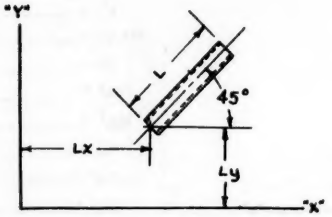
= _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 3 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L = _____
L^2 = _____
$.354L^2$ = _____
$.707L^2$ = _____
L^3 = _____
$1/6\ L^3$ = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

As = L = _____

Ax = $LLx + .354L^2$

= _____ + _____ = _____

Ay = $LLy + .354L^2$

= _____ + _____ = _____

$Ax^2 = L\,Lx^2 + .707L^2Lx + 1/6\,L^3$

= _____ + _____ + _____ = _____

$Ay^2 = L\,Ly^2 + .707L^2Ly + 1/6\,L^3$

= _____ + _____ + _____ = _____

$Axy = LLxLy + .354L^2Lx + .354L^2Ly + 1/6\,L^3$

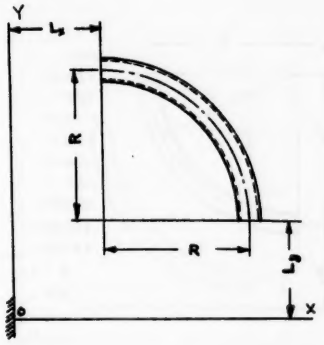
= _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 4 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ CHECKED BY _____ DATE _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
1.571KR = _____
KR^2 = _____
$2KR^2$ = _____
KR^3 = _____
.785KR^3 = _____
.50KR^3 = _____
R = _____
R^2 = _____
Lx = _____
Ly = _____
Lx^2 = _____
Ly^2 = _____
LxLy = _____

$A_s = 1.571KR =$ ___ ___ ___ ___ ___ ___

$A_x = 1.571KRL_x + KR^2 =$ ___ ___ + ___ ___ = ___ ___

$A_y = 1.571KRL_y + KR^2 =$ ___ ___ + ___ ___ = ___ ___ ___

$A_{x^2} = 1.571KRL_x^2 + .785KR^3 + 2KR^2L_x$

= ___ ___ + ___ ___ + ___ ___ ___ = ___ ___

$A_{y^2} = 1.571KRL_y^2 + .785KR^3 + 2KR^2L_y$

= ___ ___ + ___ ___ + ___ ___ ___ = ___ ___ ___

$A_{xy} = 1.571KRL_xL_y + .50KR^3 + KR^2L_x + KR^2L_y$

= ___ ___ + ___ ___ + ___ ___ + ___ ___ = ___ ___

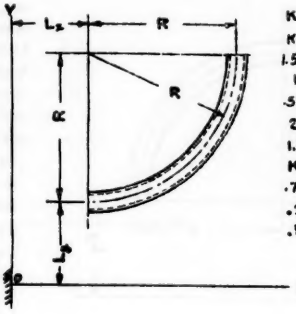
GROUP Nº ___ FORM Nº 5 SHEET Nº ___

PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
1.571KR = _____
KR² = _____
.571KR² = _____
2 KR² = _____
1.142KR² = _____
KR³ = _____
.785KR³ = _____
.355KR³ = _____
.50 KR³ = _____
R = _____
R² = _____
Lx = _____
Lx² = _____
Ly = _____
Ly² = _____
Lxy = _____

As = 1.571KR = _____

Ax = 1.571KRLx + KR² = _____ + _____ = _____

Ay = 1.571KRLy + .571KR² = _____ + _____ = _____

Ax² = 1.571KRLx² + .785KR³ + 2KR²Lx
= _____ + _____ + _____ = _____

Ay² = 1.571KRLy² + .355KR³ + 1.142KR²Ly
= _____ + _____ + _____ = _____

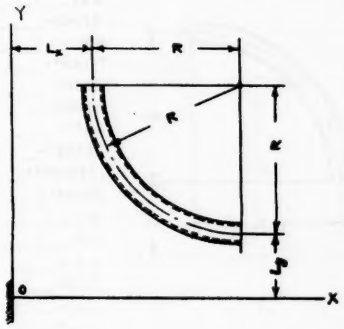
Axy = 1.571KRLx Ly + ½KR³ + .571KR²Lx + KR²Ly
= _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 6 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
1.571KR = _____
KR^2 = _____
.571 KR^2 = _____
1.142 KR^2 = _____
KR^3 = _____
.355 KR^3 = _____
.071 KR^3 = _____
R = _____
R^2 = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

A_s = 1.571KR + _____

A_x = 1.571KRL_x + .571KR^2 + _____ = _____

A_y = 1.571KRL_y + .571KR^2 + _____ = _____

A_x^2 = 1.571KRL_x^2 + .355 KR^3 + 1.142KR^2L_x

A_x^2 = _____ + _____ + _____ = _____

A_y^2 = 1.571KRL_y^2 + .355 KR^3 + 1.142KR^2L_y

A_y^2 = _____ + _____ + _____ = _____

A_{xy} = 1.571KRL_xL_y + .071KR^3 + .571KR^2L_x + .571KR^2L_y

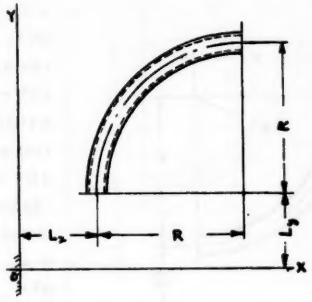
A_{xy} = _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 7 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
1.571KR = _____
KR^2 = _____
.571KR^2 = _____
2KR^2 = _____
1.142KR^2 = _____
KR^3 = _____
.785KR^3 = _____
.355KR^3 = _____
.50KR^3 = _____

R = _____
R^2 = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

As = 1.571KR = ____ ____ ____ ____ ____ = ____ ____ ____

Ax = 1.571KRLx + .571KR^2 = ____ ____ + ____ ____ = ____ ____

Ay = 1.571KRLy + KR^2 = ____ ____ + ____ ____ = ____ ____

Ax^2 = 1.571KRLx^2 + .355KR^3 + 1.142KR^2Lx

= ____ ____ + ____ ____ + ____ ____ ____ = ____ ____

Ay^2 = 1.571KRLy^2 + .785KR^3 + 2KR^2Ly

= ____ ____ + ____ ____ + ____ ____ = ____ ____

Axy = 1.571KRLxLy + .50KR^3 + .571KR^2Ly + KR^2Lx

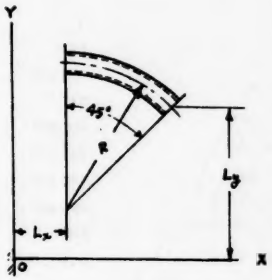
= ____ ____ + ____ ____ + ____ ____ + ____ ____ = ____ ____ ____

GROUP Nº ____ FORM Nº 8 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
$.785KR$ = _____
KR^2 = _____
$.293KR^2$ = _____
$.152KR^2$ = _____
$.586KR^2$ = _____
$.304KR^2$ = _____
KR^3 = _____
$.035KR^3$ = _____
$.143KR^3$ = _____
$.043KR^3$ = _____

R = _____ L_x = _____ L_y = _____
R^2 = _____ L_x^2 = _____ L_y^2 = _____
R^3 = _____
$L_x L_y$ = _____

$A_s = .785KR$ = _____

$A_x = .785KRL_x + .293KR^2$ = _____ + _____ = _____

$A_y = .785KRL_y + .152KR^2$ = _____ + _____ = _____

$A_x^2 = .785KRL_x^2 + .143KR^3 + .586KR^2L_x$

A_x^2 = _____ + _____ + _____ = _____

$A_y^2 = .785KRL_y^2 + .035KR^3 + .304KR^2L_y$

A_y^2 = _____ + _____ + _____ = _____

$A_{xy} = .785KRL_xL_y + .043KR^3 + .152KR^2L_x + .293KR^2L_y$

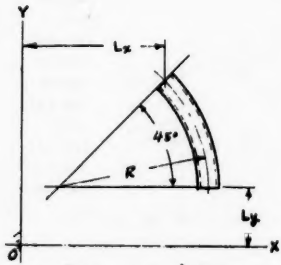
A_{xy} = _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 9 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
.785 KR = _____
KR^2 = _____
.152 KR^2 = _____
.304 KR^2 = _____
.293 KR^2 = _____
.586 KR^2 = _____
KR^3 = _____
.035 KR^3 = _____
.143 KR^3 = _____
.043 KR^3 = _____

R = _____ L_x = _____ L_y = _____
R^2 = _____ L_x^2 = _____ L_y^2 = _____
R^3 = _____ L_xL_y = _____

A_s = .785 KR = _____

A_x = .785 KRL_x + .152 KR^2 = _____

A_y = .785 KRL_y + .293 KR^2 = _____

A_x^2 = .785 KRL_x^2 + .035 KR^3 + .304 KR^2L_x

A_x^2 = _____ + _____ + _____ = _____

A_y^2 = .785 KRL_y^2 + .143 KR^3 + .586 KR^2L_y =

A_y^2 = _____ + _____ + _____ = _____

A_{xy} = .785 KRL_xL_y + .043 KR^3 + .293 KR^2L_x + .152 KR^2L_y

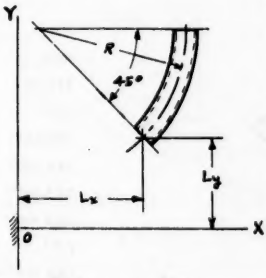
A_{xy} = _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 19 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
.785 KR = _____
KR^2 = _____
.152 KR^2 = _____
.262 KR^2 = _____
.304 KR^2 = _____
.524 KR^2 = _____
KR^3 = _____
.035 KR^3 = _____
.121 KR^3 = _____
.064 KR^3 = _____

R = _____ L_x = _____ L_y = _____
R^2 = _____ L_x^2 = _____ L_y^2 = _____
R^3 = _____ $L_x L_y$ = _____

A_s = .785 KR = _____

A_x = .785 KRL_x + .152 KR^2 = _____ + _____ = _____

A_y = .785 KRL_y + .262 KR^2 = _____ + _____ = _____

A_x^2 = .788 KRL_x^2 + .038 KR^3 + .304 KR^2L_x

A_x^2 = _____ + _____ + _____ = _____

A_y^2 = .785 KRL_y^2 + .121 KR^3 + .524 KR^2L_y

A_y^2 = _____ + _____ + _____ = _____

A_{xy} = .785 KRL_xL_y + .064 KR^3 + .262 KR^2L_x + .152 KR^2L_y

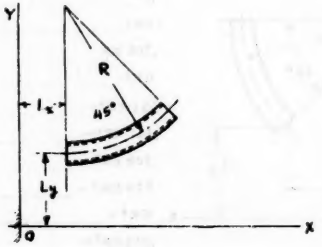
A_{xy} = _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 11 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
.785 KR = _____
KR^2 = _____
.078 KR^2 = _____
.156 KR^2 = _____
.293 KR^2 = _____
.586 KR^2 = _____
KR^3 = _____
.143 KR^3 = _____
.014 KR^3 = _____
.043 KR^3 = _____

R = _____ L_x = _____ L_y = _____
R^2 = _____ L_x^2 = _____ L_y^2 = _____
R^3 = _____ $L_x L_y$ = _____

A_s = .785 KR _____ = _____

A_x = .785 KRL_x + .293 KR^2 = _____ + _____ = _____

A_y = .785 KRL_y + .078 KR^2 = _____ + _____ = _____

A_x^2 = .785 KRL_x^2 + .143 KR^3 + .586 KR^2L_x

A_x^2 = _____ + _____ + _____ = _____

A_y^2 = .785 KRL_y^2 + .014 KR^3 + .156 KR^2L_y

A_y^2 = _____ + _____ + _____ = _____

A_{xy} = .785 KRL_xL_y + .043 KR^3 + .078 KR^2L_x + .293 KR^2L_y

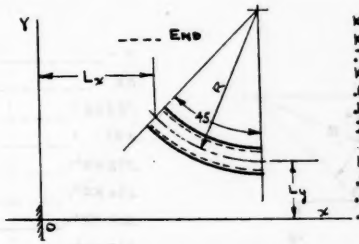
A_{xy} = _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 12 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K =
KR =
.785KR =
KR^2 =
$.078KR^2$ =
$.156KR^2$ =
$2.62KR^2$ =
$.524KR^2$ =
KR^3 =
$.012KR^3$ =
$.014KR^3$ =
$.121KR^3$ =

R = _____ L_x = _____ L_y = _____

L_x^2 = _____ L_y^2 = _____ L_xL_y = _____

$A_S = .785KR^2$ = _____ = _____

$A_x = .785KRL_x + .262KR^2$ = _____ + _____ = _____

$A_y = .785KRL_y + .078KR^2$ = _____ + _____ = _____

$A_{x^2} = .785KRL_x^2 + .121KR^3 + .524KR^2L_x$

= _____ + _____ + _____ = _____

$A_{y^2} = .785KRL_y^2 + .014KR^3 + .156KR^2L_y$

= _____ + _____ + _____ = _____

$A_{xy} = .785KRL_xL_y + .262KR^2L_y + .078KR^2L_x + .012KR^3$

= _____ + _____ + _____ ± _____ = _____

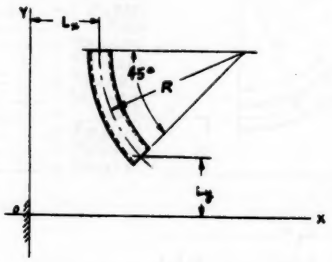
GROUP Nº ___ FORM Nº 13 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____

KR = _____

.785 KR = _____

KR^2 = _____

.078 KR^2 = _____

.156 KR^2 = _____

.262 KR^2 = _____

.524 KR^2 = _____

KR^3 = _____

.012 KR^3 = _____

.014 KR^3 = _____

.121 KR^3 = _____

R = _____ L_x = _____ L_y = _____

R^2 = _____ L_x^2 = _____ L_y^2 = _____

R^3 = _____ L_xL_y = _____

A_s = .785 KR = _____ = _____

A_x = .785 KRL_x + .078 KR^2 = _____ + _____ = _____

A_y = .785 KRL_y + .262 KR^2 = _____ + _____ = _____

A_x^2 = .785 KRL_x^2 + .014 KR^3 + .156 KR^2L_x

= _____ + _____ + _____ = _____

A_y^2 = .785 KRL_y^2 + .121 KR^3 + .524 KR^2L_y

= _____ + _____ + _____ = _____

A_{xy} = .785 KRL_xL_y + .012 KR^3 + .262 KR^2L_x + .078 KR^2L_y

= _____ + _____ + _____ + _____ = _____

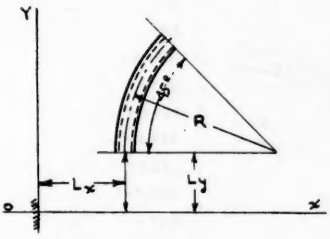
GROUP Nº ___ FORM Nº 14 SHEET Nº _____

PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
.785 KR = _____
KR^2 = _____
$.078KR^2$ = _____
$.156KR^2$ = _____
$.293KR^2$ = _____
$.586KR^2$ = _____
KR^3 = _____
$.014KR^3$ = _____
$.043KR^3$ = _____
$.143KR^3$ = _____

R = _____ L_x = _____ L_y = _____

L_x^2 = _____ L_y^2 = _____ L_xL_y = _____

A_s = .785 KR = _____ = _____

A_x = $.785KRL_x + .078KR^2$ = _____ + _____ = _____

A_y = $.785KRL_y + .293KR^2$ = _____ + _____ = _____

A_{x^2} = $.785KRL_x^2 + .014KR^3 + .156KR^2L_x$

= _____ + _____ + _____ = _____

A_{y^2} = $.785KRL_y^2 + .143KR^3 + .586KR^2L_y$

= _____ + _____ + _____ = _____

A_{xy} = $.785KRL_xL_y + .078KR^2L_y + .293KR^2L_x + .043KR^3$

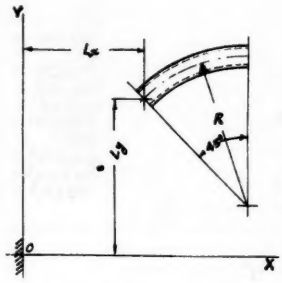
= _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 15 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



K = _____
KR = _____
.785 KR = _____
KR^2 = _____
.262 KR^2 = _____
.524 KR^2 = _____
.152 KR^2 = _____
.304 KR^2 = _____
KR^3 = _____
.121 KR^3 = _____
.035 KR^3 = _____
.064 KR^3 = _____

R = _____ L_x = _____ L_y = _____
R^2 = _____ L_x^2 = _____ L_y^2 = _____
$L_x L_y$ = _____

A_s = .785 KR = _____

A_x = .785 KRL_x + .262 KR^2 = _____ + _____ = _____

A_y = .785 KRL_y + .152 KR^2 = _____ + _____ = _____

A_x^2 = .785 KRL_x^2 + .121 KR^3 + .524 KR^2L_x
= _____ + _____ + _____ = _____

A_y^2 = .785 KRL_y^2 + .035 KR^3 + .304 KR^2L_y
= _____ + _____ + _____ = _____

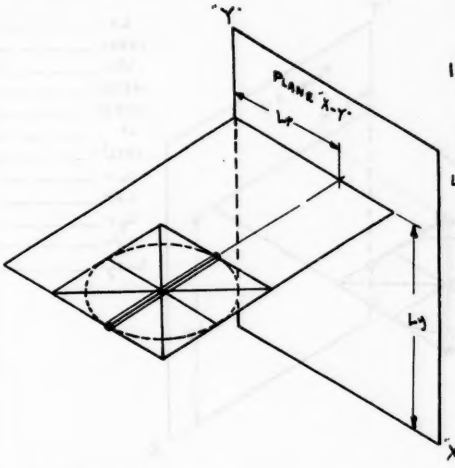
A_{xy} = .785 KRL_xL_y + .064 KR^3 + .152 KR^2L_x + .262 KR^2L_y
= _____ + _____ + _____ + _____ = _____

GROUP Nº ___ FORM Nº 16 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L= _____
1.3L= _____
Lx= _____
Lx²= _____
Ly= _____
Ly²= _____
LxLy= _____

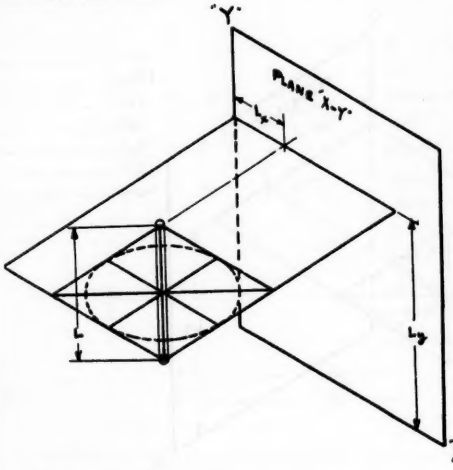
A_s= 1.3 L = _____
A_x= 1.3 LL_x = _____
A_y= 1.3 LL_y = _____
A_{x^2}= 1.3 LL_{x^2}= _____
A_{y^2}= 1.3 LL_{y^2}= _____
A_{xy}= 1.3 L$L_x L_y$= _____

GROUP Nº ___ FORM Nº 17 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



$L =$ _____
$1.15L =$ _____
$L^2 =$ _____
$.407L^2 =$ _____
$.813L^2 =$ _____
$L^3 =$ _____
$.192L^3 =$ _____
$L_x =$ _____
$L_x^2 =$ _____
$L_y =$ _____
$L_y^2 =$ _____
$L_x L_y =$ _____

$A_s = 1.15L =$ _____

$A_x = 1.15LL_x + .407L^2 =$ _____ + _____ = _____

$A_y = 1.15LL_y =$ _____

$A_x^2 = 1.15LL_x^2 + .813L^2L_x + .192L^3$
$=$ _____ + _____ + _____ = _____

$A_y^2 = 1.15LL_y^2 =$ _____

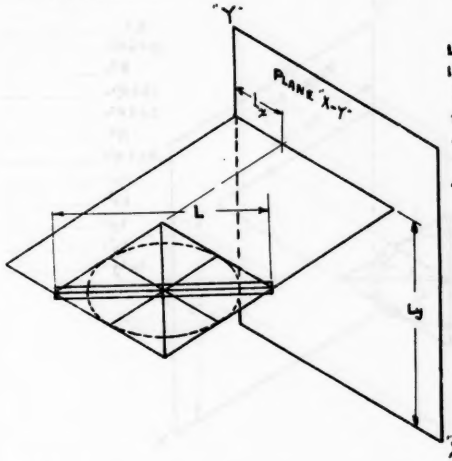
$A_{xy} = 1.15LL_xL_y + .407L^2L_y$
$=$ _____ + _____ = _____

GROUP Nº ___ FORM Nº 18 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L = _____
1.15 L = _____
L^2 = _____
.407 L^2 = _____
.813 L^2 = _____
L^3 = _____
.192 L^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____

A_s = 1.15 L _____ = _____

A_x = 1.15 L L_x + .407 L^2 = _____ + _____ = _____

A_y = 1.15 L L_y = _____

A_x^2 = 1.15 L L_x^2 + .813 $L^2 L_x$ + .192 L^3
= _____ + _____ + _____ = _____

A_y^2 = 1.15 L L_y^2 = _____

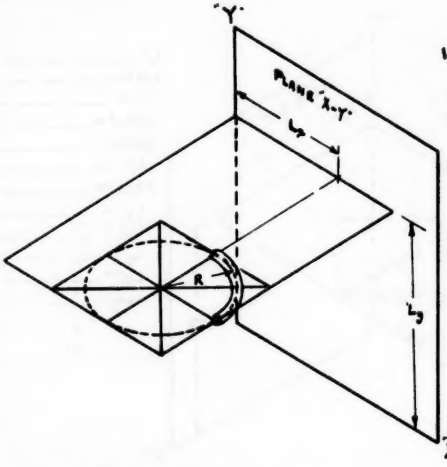
A_{xy} = 1.15 L $L_x L_y$ + .407 $L^2 L_y$
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 19 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
$1.806R$ = _____
R^2 = _____
$1.20R^2$ = _____
$2.40R^2$ = _____
R^3 = _____
$.962R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_s = $1.806R$ = _____

A_x = $1.806RL_x + 1.20R^2$ = _____ + _____ = _____

A_y = $1.806RL_y$ = _____

A_s^2 = $1.806RL_x^2 + 2.40R^2L_x + .962R^3$
= _____ + _____ + _____ = _____

A_y^2 = $1.806RL_y^2$ = _____

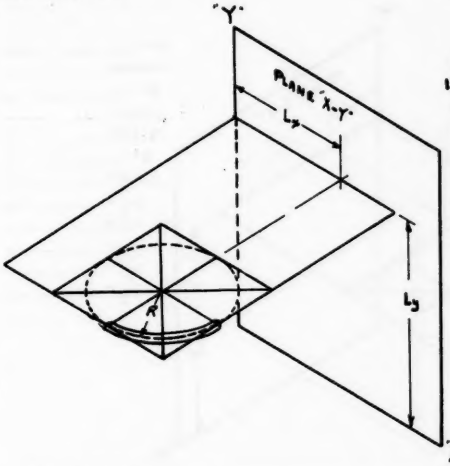
A_{xy} = $1.806RL_xL_y + 1.20R^2L_y$
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 20 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
1.806R = _____
R^2 = _____
1.20R^2 = _____
2.40R^2 = _____
R^3 = _____
.962R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_xL_y^2$ = _____

A_s = 1.806 R _____ = _____

A_x = 1.806 RL_x + 1.20 R^2 = _____ + _____ = _____

A_y = 1.806 RL_y = _____

A_x^2 = 1.806 RL_x^2 + 2.40 R^2L_x + .962 R^3
= _____ + _____ + _____ = _____

A_y^2 = 1.806 RL_y^2
= _____

A_{xy} = 1.806 RL_xL_y + 1.20 R^2L_y
= _____ + _____ = _____

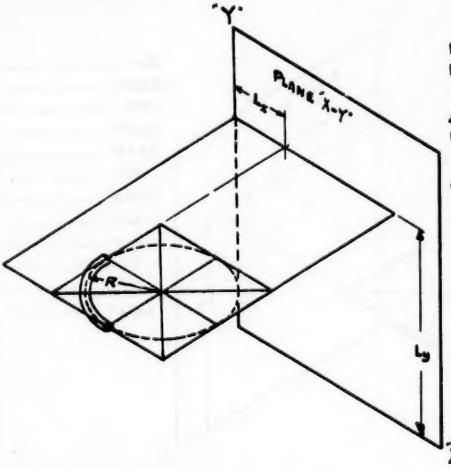
GROUP Nº ___ FORM Nº 21 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = ___ ___ ___
1.806R = ___ ___
R^2 = ___ ___ ___
.607 R^2 = ___ ___
1.212 R^2 = ___ ___ ___
R^3 = ___ ___ ___
.368 R^3 = ___ ___
L_x = ___ ___ ___
L_x^2 = ___ ___
L_y = ___ ___ ___
L_y^2 = ___ ___
$L_x L_y$ = ___ ___ ___

A_s = 1.806 R ___ ___ ___ ___ ___ ___ = ___ ___ ___

A_x = 1.806 RL_x + .607 R^2 = ___ ___ + ___ ___ ___ = ___ ___ ___

A_y = 1.806 RL_y = ___ ___ ___ ___ ___ ___ ___ ___

A_x^2 = 1.806 $R L_x^2$ + 1.212 $R^2 L_x$ + .368 R^3

= ___ ___ + ___ ___ + ___ ___ ___ ___ = ___ ___ ___

A_y^2 = 1.806 RL_y^2

= ___ ___ ___ ___ ___ ___ ___ ___ ___

A_{xy} = 1.806 RL_xL_y + .607 R^2L_y

= ___ ___ + ___ ___ ___ ___ ___ = ___ ___

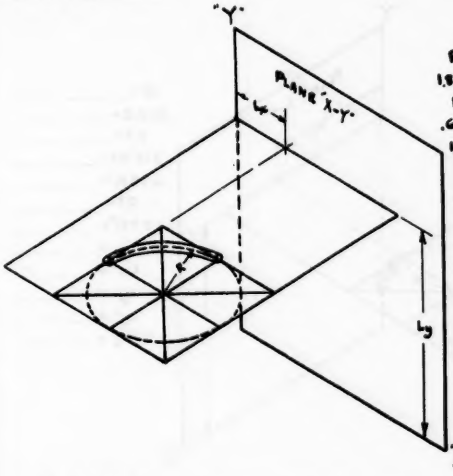
GROUP Nº ___ ___ FORM Nº 22 SHEET Nº ___ ___ ___

PAGE Nº ___ ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R= ___ ___ ___
1.806R= ___ ___ ___
R^2 = ___ ___ ___
.607R^2= ___ ___ ___
1.212R^2= ___ ___ ___
R^3= ___ ___
.368R^3= ___ ___ ___
L_x= ___ ___
L_x^2= ___ ___ ___
L_y= ___ ___
L_y^2= ___ ___ ___
L_xL_y= ___ ___

A_s= 1.806 R = ___ ___ ___ ___

A_x= 1.806 RL_x + .607 R^2 = ___ ___ + ___ ___ = ___ ___ ___

A_y= 1.806 RL_y = ___ ___ ___ ___ ___ ___

A_x^2= 1.806 RL_x^2 + 1.212 R^2L_x + .368 R^3

= ___ ___ + ___ ___ + ___ ___ = ___ ___ ___

A_y^2= 1.806 RL_y^2

= ___ ___ ___ ___ ___ ___ ___ ___

A_{xy}= 1.806 RL_xL_y + .607 R^2L_y

= ___ ___ + ___ ___ ___ = ___ ___ ___

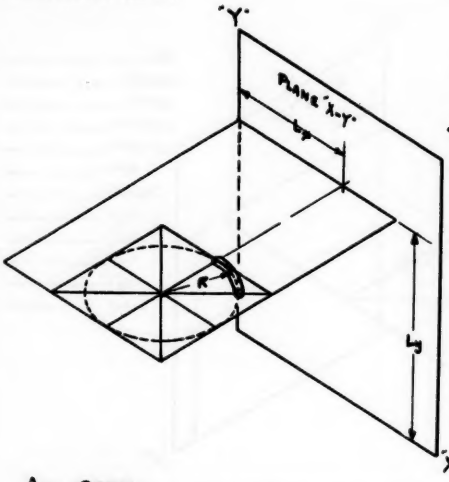
GROUP Nº ___ ___ FORM Nº 23 SHEET Nº ___ ___
PAGE Nº ___ ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____

.828 R = _____

R^2 = _____

.315 R^2 = _____

.633 R^2 = _____

R^3 = _____

.157 R^3 = _____

L_x = _____

L_x^2 = _____

L_y = _____

L_y^2 = _____

$L_x L_y$ = _____

A_0 = .828 R = _____

A_x = .828 R L_x + .315 R^2 = _____ + _____ = _____

A_y = .828 R L_y = _____

A_x^2 = .828 R L_x^2 + .633 $R^2 L_x$ + .157 R^3

= _____ + _____ + _____ = _____

A_y^2 = .828 R L_y^2

= _____

A_{xy} = .828 R $L_x L_y$ + .315 $R^2 L_y$

= _____ + _____ = _____

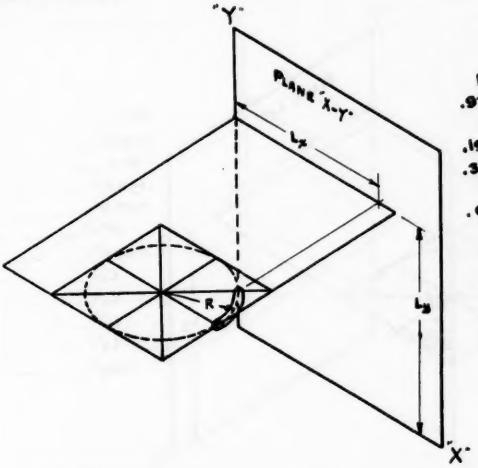
GROUP Nº ___ FORM Nº 24 SHEET Nº _____

PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
$.978R$ = _____
R^2 = _____
$.193R^2$ = _____
$.382R^2$ = _____
R^3 = _____
$.048R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

$A_s = .978R$ = _____

$A_x = .978RL_x + .193R^2$ = _____ + _____ = _____

$A_y = .978RL_y$ = _____

$A_x^2 = .978RL_x^2 + .382R^2L_x + .048R^3$
= _____ + _____ + _____ = _____

$A_y^2 = .978RL_y^2$
= _____

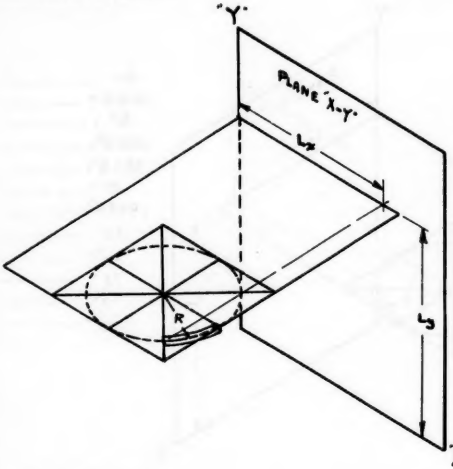
$A_{xy} = .978RL_xL_y + .193R^2L_y$
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 25 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
$.978R$ = _____
R^2 = _____
$.193R^2$ = _____
$.382R^2$ = _____
R^3 = _____
$.048R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_s = $.978R$ = _____

A_x = $.978RL_x + .193R^2$ = _____ + _____ = _____

A_y = $.978RL_y$ = _____

A_x^2 = $.978RL_x^2 + .382R^2L_x + .048R^3$
= _____ + _____ + _____ = _____

A_y^2 = $.978RL_y^2$
= _____ _____

A_{xy} = $.978RL_xL_y + .193R^2L_y$
= _____ + _____ = _____

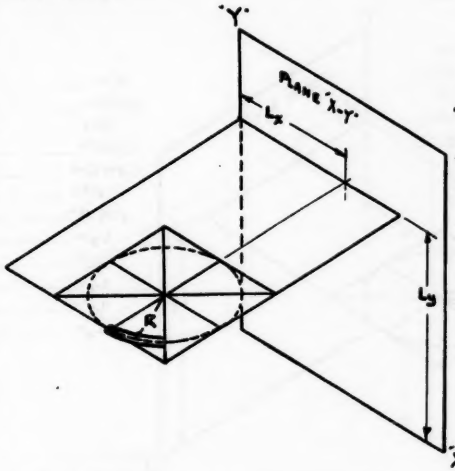
GROUP Nº ___ FORM Nº 26 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = ___ ___
.828R = ___ ___
R^2 = ___ ___
$.315R^2$ = ___ ___
$.633R^2$ = ___ ___
R^3 = ___ ___
$.157R^3$ = ___ ___
Lx = ___ ___
Lx^2 = ___ ___
Ly = ___ ___
Ly^2 = ___ ___
LxLy = ___ ___

A_s = .828R = ___ ___ ___ ___ ___ ___ ___

A_x = .828R Lx + .315 R^2 = ___ + ___ = ___ ___

A_y = .828R Ly = ___ ___ ___ ___ ___

A_{x^2} = .828R Lx^2 + .633 R^2 Lx + .157 R^3

= ___ ___ + ___ ___ + ___ ___ = ___ ___

A_{y^2} = .828R Ly^2

= ___ ___ ___ ___ ___ ___ ___

A_{xy} = .828R LxLy + .315 R^2 Ly

= ___ ___ + ___ ___ ___ = ___ ___

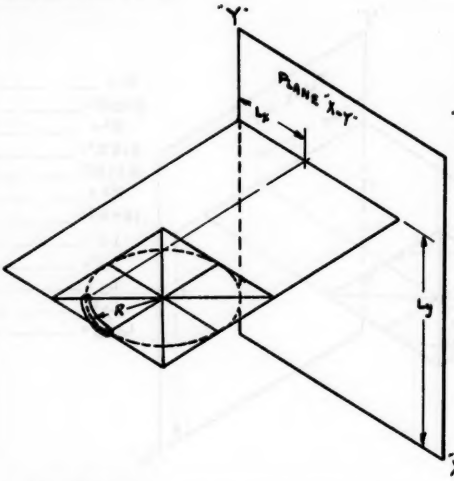
GROUP Nº ___ FORM Nº 27 SHEET Nº ___

PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
.828R = _____
R^2 = _____
$.270R^2$ = _____
$.539R^2$ = _____
R^3 = _____
$.124R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

$A_s = .828R$ = _____

$A_x = .828RL_x + .270R^2$ = _____ + _____ = _____

$A_y = .828RL_y$ = _____

$A_x^2 = .828RL_x^2 + .539R^2L_x + .124R^3$
= _____ + _____ + _____ = _____

$A_y^2 = .828RL_y^2$
= _____

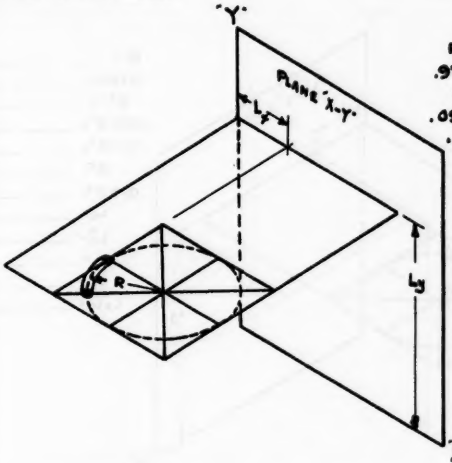
$A_{xy} = .828RL_xL_y + .270R^2L_y$
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 28 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____
CHECKED BY _____ DATE _____

SECTION Nº ___ ___ PLANE



R = _____
.978 R = _____
R^2 = _____
.095 R^2 = _____
.191 R^2 = _____
R^3 = _____
.019 R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____

A_s = .978 R = _____

A_x = .978 R L_x + .095 R^2 = _____ + _____ = _____

A_y = .978 R L_y = _____

A_x^2 = .978 R L_x^2 + .191 $R^2 L_x$ + .019 R^3
= _____ + _____ + _____ = _____

A_y^2 = .978 R L_y^2
= _____

A_{xy} = .978 R $L_x L_y$ + .095 $R^2 L_y$
= _____ + _____ = _____

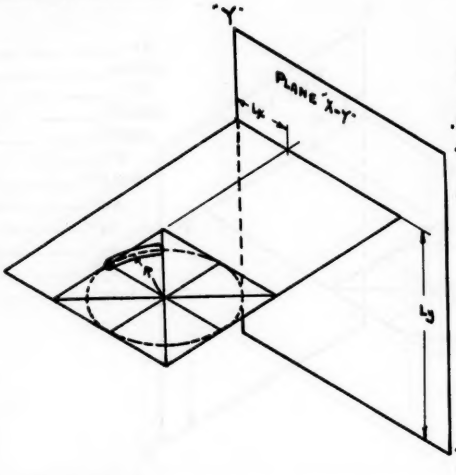
GROUP Nº ___ FORM Nº 29 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = ___ _____

$.978R$ = _____ ___

R^2 = _____ ___

$.095R^2$ = _____ ___

$.191R^2$ = _____ ___

R^3 = _____ ___

$.019R^3$ = _____ ___

L_x = _____ ___

L_x^2 = _____ ___

L_y = _____ ___

L_y^2 = _____ ___

L_xL_y = _____ ___

A_s = $.978R$ = _____ _____ _____ _____ _____ _____

A_x = $.978RL_x + .095R^2$ = _____ _____ + _____ _____ = _____ _____

A_y = $.978RL_y$ = _____ _____ _____ _____ _____

A_x^2 = $.978RL_x^2 + .191R^2L_x + .019R^3$

= _____ _____ + _____ _____ + _____ _____ = _____ _____ _____

A_y^2 = $.978RL_y^2$

= _____ _____ _____ _____ _____ _____

A_{xy} = $.978RL_xL_y + .095R^2L_y$

= _____ _____ + _____ _____ = _____ _____ _____ _____

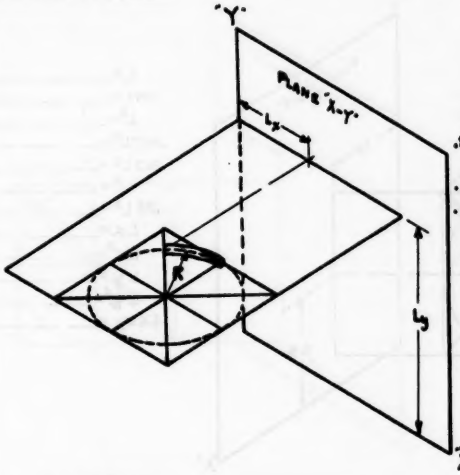
GROUP Nº ___ ___ FORM Nº 30 SHEET Nº _____

PAGE Nº ___ ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
.828 R = _____
R^2 = _____
.270 R^2 = _____
.539 R^2 = _____
R^3 = _____
.124 R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____

A_s = .828 R = _____

A_x = .828 RL_x + .270 R^2 = _____

A_y = .828 RL_y = _____

A_x^2 = .828 RL_x^2 + .539 R^2L_x + .124 R^3

= _____ + _____ + _____ = _____

A_y^2 = .828 RL_y^2

= _____

A_{xy} = .828 RL_xL_y + .270 R^2L_y

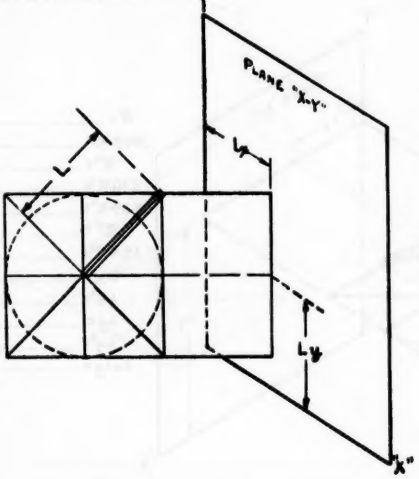
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 31 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



L = ___
1.15L = ___
L^2 = ___
$.407L^2$ = ___
$.813L^2$ = ___
L^3 = ___
$.192L^3$ = ___
Lx = ___
Lx^2 = ___
Ly = ___
Ly^2 = ___
LxLy = ___

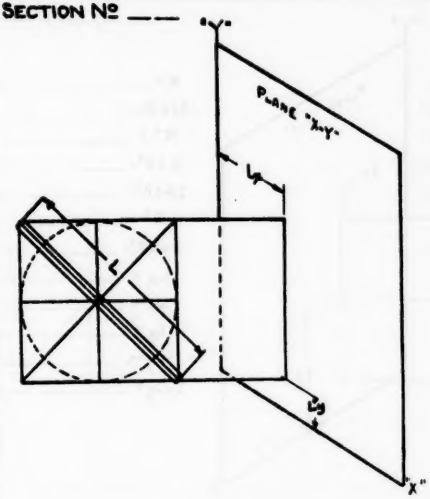
$A_z = 1.15L =$ _____
$A_x = 1.15LLx =$ _____
$A_y = 1.15LLy + .407L^2 =$ _____ + _____ = _____
$A_x^2 = 1.15LLx^2$
= _____ = _____
$A_y^2 = 1.15LLy^2 + .813L^2Ly + .192L^3$
= _____ + _____ + _____ = _____
$A_{xy} = 1.15LLxLy + .407L^2Lx$
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 32 SHEET Nº ___
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



L = _____
1.15L = _____
L^2 = _____
$.407L^2$ = _____
$.813L^2$ = _____
L^3 = _____
$.192L^3$ = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

A_s = 1.15L = _____

A_x = 1.15LLx = _____

A_y = 1.15LLy + $.407L^2$ = _____ + _____ = _____

A_x^2 = 1.15LLx²

= _____

A_y^2 = 1.15LLy² + $.813L^2$Ly + $.192L^3$

= _____ + _____ + _____ = _____

A_{xy} = 1.15LLxLy + $.407L^2$Lx

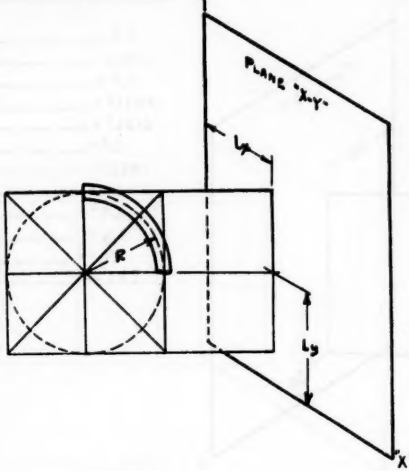
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 33 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = _____
$1.806R$ = _____
R^2 = _____
$1.20R^2$ = _____
$2.40R^2$ = _____
R^3 = _____
$.962R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_s: $1.806\,R$ = _____

A_x: $1.806\,RL_x$ = _____

A_y: $1.806\,RL_y + 1.20R^2$ = _____ + _____ = _____

A_{x^2}: $1.806\,RL_x^2$ = _____

=

A_{y^2}: $1.806\,RL_y^2 + 2.40\,R^2L_y + .962\,R^3$

= _____ + _____ + _____ = _____

A_{xy}: $1.806\,RL_xL_y + 1.20R^2L_x$

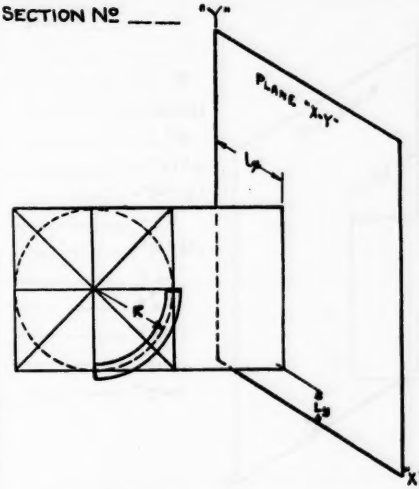
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 34 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ____ "Y" ____ PLANE



R = _____ _____
$1.806R$ = _____ _____
R^2 = _____ _____
$.607R^2$ = _____ _____
$1.212R^2$ = _____ _____ _____
R^3 = _____ _____ _____
$.368R^3$ = _____ _____ _____
L_x = _____ _____ _____
L_x^2 = _____ _____ _____
L_y = _____ _____ _____
L_y^2 = _____ _____ _____
L_xL_y = _____ _____

A_z = $1.806R$ = _____ _____ _____ _____

A_x = $1.806RL_x$ = _____ _____ _____ _____

A_y = $1.806RL_y + .607R^2$ = ____ ____ + ____ ____ = ____ ____

A_x^2 = $1.806RL_x^2$

= _____ _____ = _____ _____ _____ _____ _____

A_y^2 = $1.806RL_y^2 + 1.212R^2L_y + .368R^3$

= _____ _____ + _____ _____ + _____ _____ = _____ _____ _____

M_{xy} = $1.806RL_xL_y + .607R^2L_x$

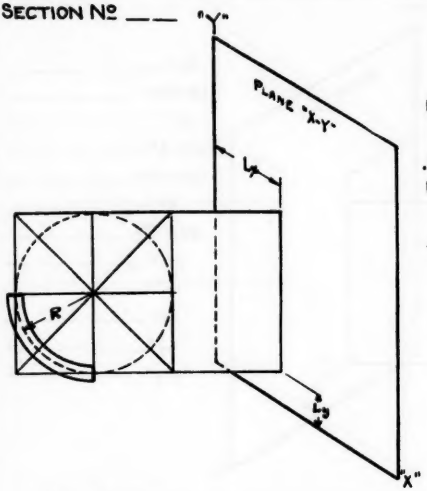
= _____ _____ + _____ _____ = _____ _____ _____ _____ _____

GROUP Nº ____ FORM Nº 35 SHEET Nº _____
PAGE Nº ____ SHEET ____ OF ____

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
1.806R = _____
R^2 = _____
.607R^2 = _____
1.212R^2 = _____
R^3 = _____
.368R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_s = 1.806R = _____

A_x = 1.806RL_x = _____

A_T = 1.806RL_y + .607R^2 = _____ + _____ = _____

A_x^2 = 1.806RL_x^2
= _____ = _____

A_T^2 = 1.806RL_y^2 + 1.212R^2L_y + .368R^3
= _____ + _____ + _____ = _____

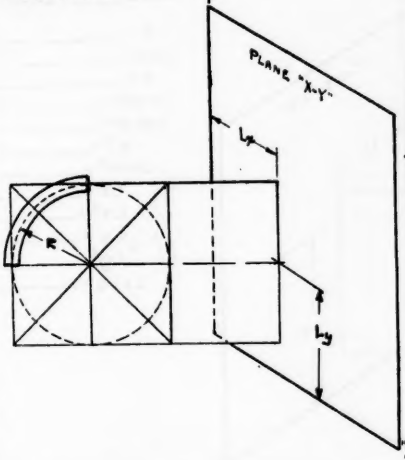
A_{xy} = 1.806RL_xL_y + .607R^2L_x
= _____ + _____ - _____

GROUP Nº ___ FORM Nº 36 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = _____
$1.806R$ = _____
R^2 = _____
$1.20R^2$ = _____
$2.40R^2$ = _____
R^3 = _____
$.962R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_S: $1.806R$ = _____

A_x: $1.806RL_x$ = _____

A_Y: $1.806RL_y + 1.20R^2$ = _____ + _____ = _____

A_x^2: $1.806RL_x^2$ =

= _____ = _____

A_Y^2: $1.806RL_y^2 + 2.40R^2L_y + .962R^3$

= _____ + _____ + _____ = _____

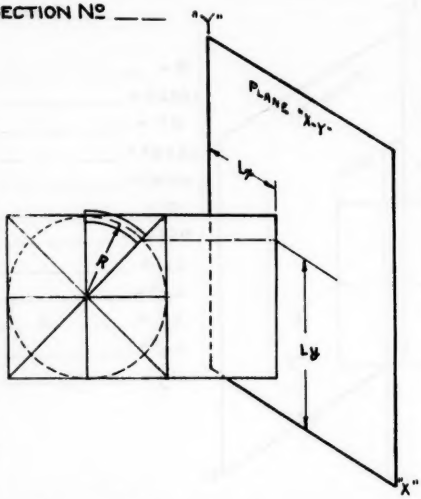
A_{xy}: $1.806RL_xL_y + 1.20R^2L_x$

= _____ + _____ = _____

GROUP Nº ___ FORM Nº 37 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ 

___ PLANE

R = _____
.978R = _____
R^2 = _____
$.193R^2$ = _____
$.382R^2$ = _____
R^3 = _____
$.048R^3$ = _____
Lx = _____
Lx^2 = _____
Ly = _____
Ly^2 = _____
LxLy = _____

A_s = .978R = _____

A_x = .978RLx = _____

A_y = .978RLy + $.193R^2$ = _____ + _____ = _____

A_{x^2} = .978RLx^2

= _____ = _____

A_{y^2} = .978RLy^2 + $.382R^2$Ly + .048R^3

= _____ + _____ + _____ = _____

A_{xy} = .978RLxLy + $.193R^2$Lx

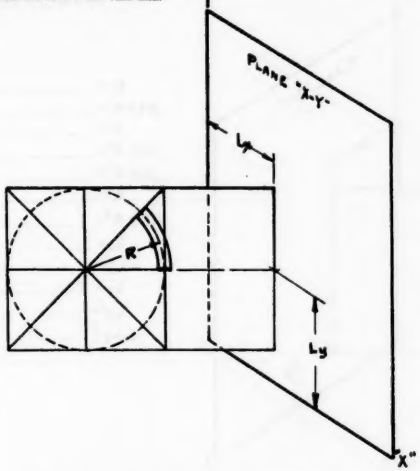
= _____ + _____ = _____

GROUP Nº ___ FORM Nº 30 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = ___
.828 R = ___
R^2 = ___
.315 R^2 = ___
.633 R^2 = ___
R^3 = ___
.157 R^3 = ___
Lx = ___
L_x^2 = ___
Ly = ___
L_y^2 = ___
LxLy = ___

A_s = .828 R = ___

A_x = .828 RL_x = ___

A_y = .828 RL_y + .315 R^2 = ___ + ___ = ___

A_x^2 = .828 RL_x^2 =

= ___ = ___

A_y^2 = .828 RL_y^2 + .633 R^2L_y + .157 R^3 =

= ___ + ___ + ___ = ___

A_{xy} = .828 RL_xL_y + .315 R^2L_x

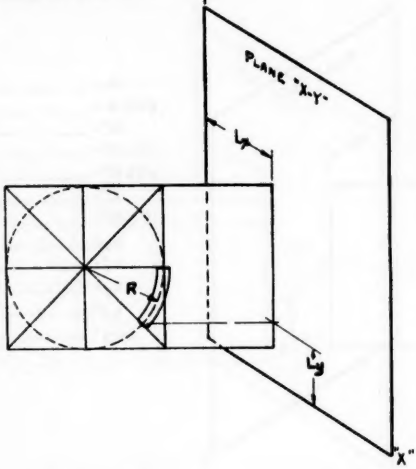
= ___ + ___ = ___

GROUP Nº ___ FORM Nº 39 SHEET Nº ___
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ____ "Y" ____ PLANE



R = _____
.828 R = _____
R^2 = _____
.270 R^2 = _____
.539 R^2 = _____
R^3 = _____
.124 R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_z = .828 R² = _____

A_x = .828 RL_x = _____

A_y = .828 RL_y + .270 R^2 = _____ + _____ = _____

A_x^2 = .828 RL_x^2 =

= _____ = _____

A_y^2 = .828 RL_y^2 + .539 R^2L_y + .124 R^3 =

= _____ + _____ + _____ = _____

A_{xy} = .828 RL_xL_y + .270 R^2L_x

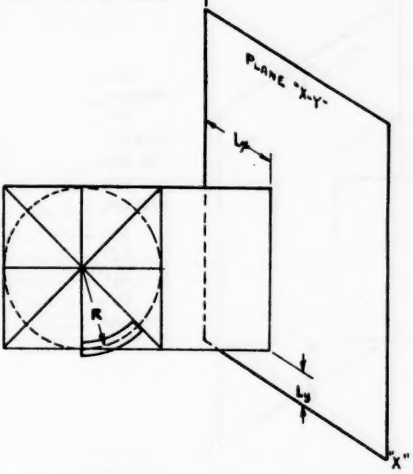
= _____ + _____ = _____.

GROUP Nº ____ FORM Nº 40 SHEET Nº _____
PAGE Nº ____ SHEET __ OF __

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = ___
.978 R = ___
R^2 = ___
.095 R^2 = ___
.191 R^2 = ___
R^3 = ___
.019 R^3 = ___
L_x = ___
L_x^2 = ___
L_y = ___
L_y^2 = ___
L_xL_y = ___

A_s = .978 R = _____

A_x = .978 RL_x = _____

A_y = .978 RL_y + .095 R^2 = ___ + ___ = _____

A_x^2 = .978 RL_x^2 =

= ___ = _____

A_y^2 = .978 RL_y^2 + .191 R^2L_y + .019 R^3

= ___ + ___ + ___ = ___

A_{xy} = .978 RL_xL_y + .095 R^2L_x

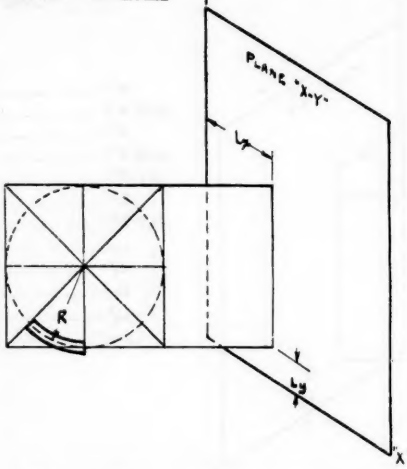
= ___ + ___ = _____

GROUP Nº ___ FORM Nº 41 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = _____
$.978\,R$ = _____
R^2 = _____
$.095\,R^2$ = _____
$.191\,R^2$ = _____
R^3 = _____
$.019\,R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
$L_x L_y$ = _____

$A_S = .978\,R =$ _____

$A_x = .978\,RL_x =$ _____

$A_T = .978\,RL_y + .095\,R^2 =$ _____ + _____ = _____

$A_x^2 = .978\,RL_x^2 =$

= _____

$A_T^2 = .978\,RL_y^2 + .191\,R^2L_y + .019\,R^3 =$

= _____ + _____ + _____ = _____

$A_{xy} = .978\,RL_xL_y + .095\,R^2L_x$

= _____ + _____ = _____

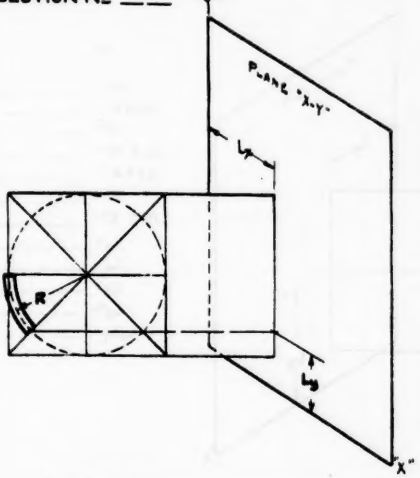
GROUP Nº ___ FORM Nº 42 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

SECTION NO ___ ___ PLANE



R = _____
$.828R$ = _____
R^2 = _____
$.270R^2$ = _____
$.539R^2$ = _____
R^3 = _____
$.124R^3$ = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

A_s = $.828R$ = _____

A_x = $.828RL_x$ = _____

A_y = $.828RL_y + .270R^2$ = _____ + _____ = _____

A_x^2 = $.828RL_x^2$

= _____ = _____

A_y^2 = $.828RL_y^2 + .539R^2L_y + .124R^3$ =

= _____ + _____ + _____ = _____

A_{xy} = $.828RL_xL_y + .270R^2L_x$

= _____ + _____ = _____

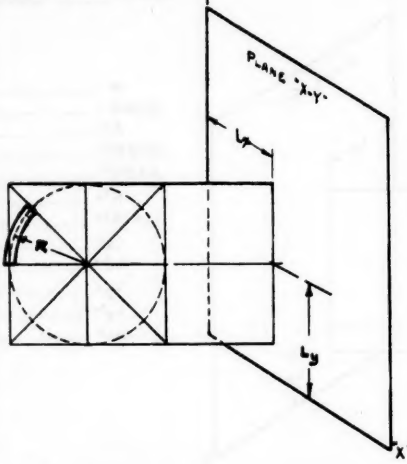
GROUP NO ___ FORM NO 43 SHEET NO _____

PAGE NO ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ "Y" ___ PLANE



R = _____
.828R = _____
R^2 = _____
.315 R^2 = _____
.633 R^2 = _____
R^3 = _____
.157 R^3 = _____
L_x = _____
L_x^2 = _____
L_y = _____
L_y^2 = _____
L_xL_y = _____

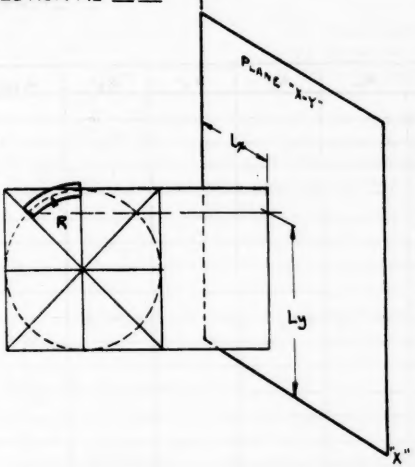
A_s = .828R = _____
A_x = .828RL_x = _____
A_Y = .828RL_y + .315 R^2 = _____ + _____ = _____
A_x^2 = .828RL_x^2
= _____ = _____
A_Y^2 = .828 RL_y^2 + .633 R^2L_y + .157 R^3 =
= _____ + _____ + _____ = _____
A_{xy} = .828 RL_xL_y + .315 R^2L_x
= _____ + _____ = _____.

GROUP Nº ___ FORM Nº 44 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION Nº ___ ___ PLANE



R = _____
.978 R = _____
R^2 = _____
.193 R^2 = _____
.382 R^2 = _____
R^3 = _____
.048 R^3 = _____
Lx = _____
L_x^2 = _____
Ly = _____
L_y^2 = _____
L_xL_y = _____

A_s = .978 R = _____

A_x = .978 RL_x = _____

A_T = .978 RL_y + .193 R^2 = _____ + _____ = _____

A_x^2 = .978 RL_x^2 = _____

= _____

A_T^2 = .978 RL_y^2 + .382 R^2L_y + .048 R^3

= _____ + _____ + _____ = _____

A_{xy} = .978 RL_xL_y + .193 R^2L_x

= _____ + _____ = _____

GROUP Nº ___ FORM Nº 45 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

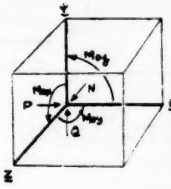
SECTION Nº ___ ___ PLANE

ELEMENT NUMBER	A_s	A_x	A_y	A_{x^2}	A_{y^2}	A_{xy}
1						
2						
3						
4						
5						
6						
7						
8						
9						
10						
11						
12						
13						
14						
15						
16						
17						
18						
19						
20						
TOTALS						
PLANE XY	A_s	A_x	A_y	A_{x^2}	A_{y^2}	A_{xy}
PLANE YZ	B_s	B_y	B_z	B_{y^2}	B_{z^2}	B_{yz}
PLANE ZX	C_s	C_z	C_x	C_{z^2}	C_{x^2}	C_{zx}

↳ ✓ CHECK PLANE USED.

GROUP Nº ___ FORM Nº 46 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

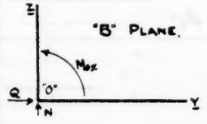
SUBJECT: GENERAL ARRANGEMENT OF PLANES FOR A THREE PLANE PIPE.



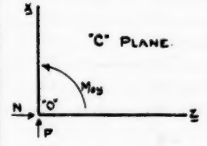
XY PLANE KNOWN AS "A" PLANE
YZ PLANE KNOWN AS "B" PLANE
ZX PLANE KNOWN AS "C" PLANE

CONSIDER EACH PLANE SEPARATELY – ASSUME LOOKING FROM INSIDE OF BOX OUT TO DETERMINE ROTATION – ORIENT INDIVIDUAL PLANES TO COINCIDE WITH DIAGRAM SO THAT EQUATIONS CAN CONFORM TO TEXT

$$M_z = M_{oz} + Py - Qx$$



$$M_x = M_{ox} + Qz - Ny$$



$$M_y = M_{oy} + Nx - Pz$$

GROUP Nº ___ FORM Nº 47 SHEET Nº ___
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: COEFFICIENTS FOR SECTION OF PIPE FOR _____

COMPUTED BY _____ DATE _____
CHECKED BY _____

SECTION NO. ___ ___ PLANE

Cx^2 =	Ax^2 =
By^2 = +	Bz^2 = +
Cx^2+By^2 =	Ax^2+Bz^2 =
Bs =	As =
By =	Ax =
$(By)^2$ =	$(Ax)^2$ =
$(By)^2/Bs$ =	$(Ax)^2/As$ =
Cs =	Bs =
Cx =	Bz =
$(Cx)^2$ =	$(Bz)^2$ =
$(Cx)^2/Cs$ = +	$(Bz)^2/Bs$ = +
$(By)^2/Bs+(Cx)^2/Cs$ = −	$(Ax)^2/As+(Bz)^2/Bs$ = −
α_1^2 =	λ_2^2 =
C_{xy} =	A_{xy} =
Cs =	As =
Cx =	Ax =
Cz =	Ay =
CxCz =	AxAy =
CxCz/Cs = −	AxAy/As = −
β_1^2 =	β_2^2 =
B_{yz} =	Ay^2 =
Bs =	Cz^2 = +
By =	Ay^2+Cz^2 =
Bz =	As =
ByBz =	Ay =
ByBz/Bs = −	$(Ay)^2$ =
α_2^2 =	$(Ay)^2/As$ =
	Cs =
	Cz =
	$(Cz)^2$ =
	$(Cz)^2/Cs$ = +
	$(Ay)^2/As+(Cz)^2/Cs$ = −
	β_3^2 =

$$+\alpha_1 N - \beta_1 P - \alpha_2 Q = Elay$$
$$-\alpha_2 N - \beta_2 P + \lambda_2 Q = Elay$$
$$-\beta_1 N + \beta_3 P - \beta_2 Q = Elax$$

GROUP NO. ___ FORM NO. 48 SHEET NO. ___
PAGE NO. ___ SHEET ___ OF ___

COMPUTATION OF STRESS IN PIPES.

TABLE FOR SELECTION OF FORMS FOR 3-PLANE PIPES

WHEN IN THE X-Y PLANE THE PIPE ELEMENT IS THAT CORRESPONDING TO THE FORM NUMBER GIVEN IN THE -A- COLUMNS, THEN THE CORRESPONDING FORMS FOR THE SAME ELEMENT FOR THE Y-Z AND Z-X PLANES WILL BE THOSE TABULATED IN THE -B- AND -C- COLUMNS, RESPECTIVELY

FORM NUMBERS					
-A- FOR X-Y PLANE	-B- FOR Y-Z PLANE	-C- FOR Z-X PLANE	-A- FOR XY PLANE	-B- FOR YZ PLANE	-C- FOR Z-X PLANE
1	17	2	26	44	9
2	1	17	27	45	10
3	19	32	28	38	11
4	18	33	29	39	12
5	21	37	30	40	13
6	22	34	31	41	14
7	23	35	32	3	19
8	20	36	33	4	18
9	26	44	34	6	22
10	27	45	35	7	23
11	28	38	36	8	20
12	29	39	37	5	21
13	30	40	38	11	28
14	31	41	39	12	29
15	24	42	40	13	30
16	25	43	41	14	31
17	2	1	42	15	24
18	33	4	43	16	25
19	32	3	44	9	26
20	36	8	45	10	27
21	37	5			
22	34	6			
23	35	7			
24	42	15			
25	43	16			

FORM Nº 49

SUBJECT Tabulation of Form Numbers and Dimensions for use with Same

COMPUTED BY _____ DATE _____
CHECKED BY _____

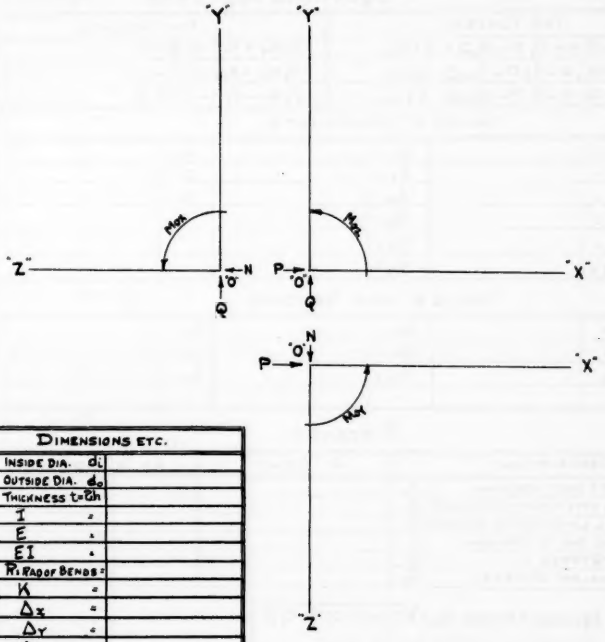
Element Number	Length "L"	Radius "R"	for "X-Y" Plane			for "Y-Z" Plane			for "Z-X" Plane		
			Form No	"X" L_x	"Y" L_y	Form No	"Y" L_x	"Z" L_y	Form No	"Z" L_x	"X" L_y

NOTE! "X" is the Dimension from Nearest End of Element to "Y-Z" Plane
"Y" " " " " " " " " " "Z-X" "
"Z" " " " " " " " " " "X-Y" "
For Form Numbers for "Y-Z" and "Z-X" Planes see Form No 49

GROUP No. _____ FORM No 50 SHEET No. _____
SHEET __ OF __

SUBJECT: DIMENSIONS FOR 3-PLANE PIPE

COMPUTED BY _____ DATE _____
CHECKED BY _____



DIMENSIONS ETC.	
INSIDE DIA. d_i	
OUTSIDE DIA. d_o	
THICKNESS $t=\frac{d_o}{2}h$	
I "	
E "	
EI "	
R₁ RAD OF BENDS "	
K "	
Δ_x "	
Δ_y "	
Δ_z "	
TEMPERATURE "	
PRESSURE	

NOTE! "O" END OF PIPE IS LOCATED AT _____

GROUP Nº ____ FORM Nº 51 SHEET Nº _____
PAGE Nº ____ SHEET __ OF __

SUBJECT: FORCES MOMENTS AND STRESSES IN 3 PLANE PIPE

COMPUTED BY _____ DATE _____
CHECKED BY _____

EQUATIONS FOR "O" END

FOR FORCES	FOR MOMENTS.
$-\beta_1 N + \beta_2 P - \beta_3 Q = EI\Delta_x$	$A_3 M_{ox} + A_1 P - A_2 Q = 0$
$-\alpha_2 N - \beta_2 P + \lambda_2 Q = EI\Delta_Y$	$B_3 M_{ox} + B_2 Q - B_Y N = 0$
$+\alpha_1 N - \beta_1 P - \alpha_2 Q = EI\Delta_z$	$C_3 M_{oy} + C_2 N - C_1 P = 0$

VALUES OF COEFFICIENTS

α_1 =		A_3 =		C_3 =	
α_2 =		A_1 =		C_2 =	
β_1 =		A_2 =		C_1 =	
β_2 =		B_3 =			
β_3 =		B_2 =			
λ_2 =		B_Y =			

FORCES AND MOMENTS

P =		M_{ox} =		M_{Ax} =	
Q =		M_{oy} =		M_{Ay} =	
N =		M_{oz} =		M_{Az} =	

STRESSES

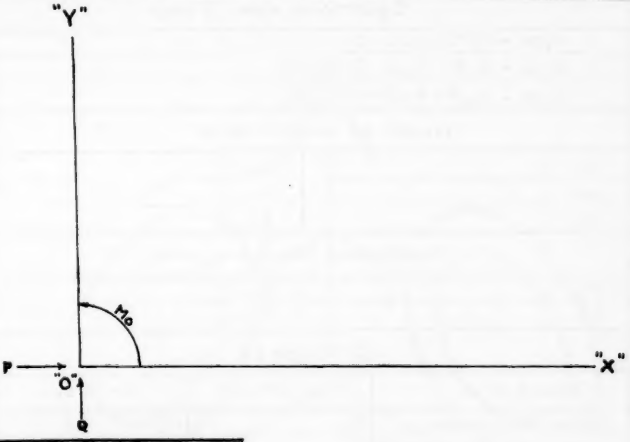
DESCRIPTION		AT "O" END	AT "A" END
STRESS DUE TO BENDING =	σ_b		
LONG'L STRESS DUE TO PRESSURE =	σ_l		
TOTAL LONGITUDINAL STRESS =	σ_0		
STRESS DUE TO TORSION =	T		
HOOP STRESS =	σ_t		
EQUIVALENT STRESS =	σ_{eq}		

EQUIVALENT STRESS $\sigma_{eq} = \sqrt{\sigma_0^2 + \sigma_t^2 + 3T^2 - \sigma_0 \sigma_t}$

GROUP No. _____ FORM No. 52 SHEET No. _____
PAGE No. _____ SHEET _____ OF _____

SUBJECT: DIMENSIONS FOR SINGLE PLANE PIPE

COMPUTED BY _____ DATE _____
CHECKED BY _____



DIMENSIONS ETC.	
INSIDE DIA. d_i	
OUTSIDE DIA d_o	
THICKNESS $t = \frac{d_o - d_i}{2}$	
I =	
E =	
ET =	
R = RAD. OF BENDS =	
K =	
Δx =	
Δy =	
TEMPERATURE =	
PRESSURE =	

NOTE! "O" END OF PIPE IS LOCATED AT _____

GROUP Nº ____ FORM Nº 53 SHEET Nº _____
PAGE Nº ____ SHEET ____ OF ____

SUBJECT: FORCES MOMENTS AND STRESSES IN SINGLE PLANE PIPE

COMPUTED BY _____ DATE _____
CHECKED BY _____

EQUATIONS FOR "O" END

$$A_s M_o + A_y P - A_x Q = 0$$

$$A_y M_o + A_{y'} P - A_{xy} Q = EI\Delta_x$$

$$A_x M_o + A_{xy} P - A_{x'} Q = -EI\Delta_y$$

VALUES OF COEFFICIENTS

$A_s =$		$A_{x'} =$	
$A_x =$		$A_{y'} =$	
$A_y =$		A_{xy}	

FORCES AND MOMENTS

$P =$		$M_o =$	
$Q =$		$M_A =$	

STRESSES

DESCRIPTION	AT "O" END	AT "A" END
STRESS DUE TO BENDING	σ_{Ob}	σ_{Ab}
LONG'L STRESS DUE TO PRESSURE	σ_s	σ_s
TOTAL LONGITUDINAL STRESS	σ_o	σ_A
HOOP STRESS	σ_t	σ_t
RESULTANT TENSILE STRESS	σ_{Od}	σ_{Ad}

RESULTANT TENSILE STRESS $\sigma_{Od} = \sqrt{\sigma_o^2 + \sigma_t^2}$

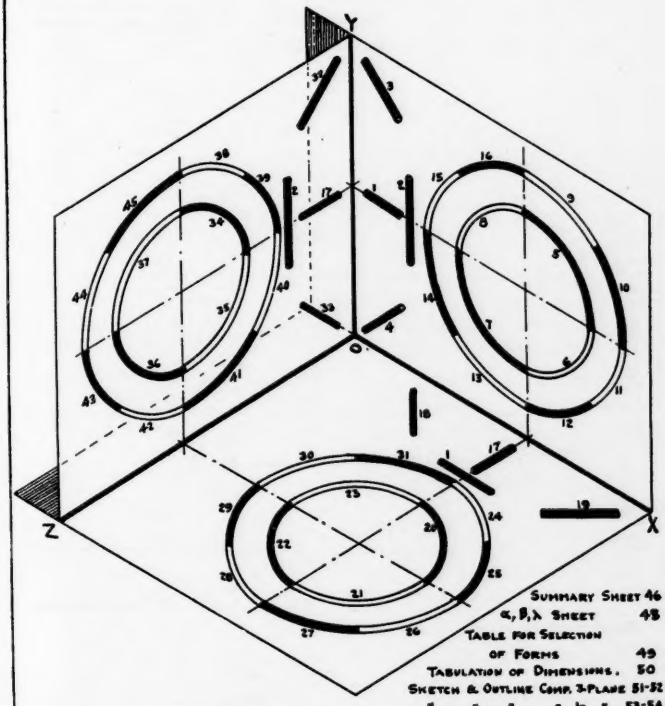
GROUP Nº ___ FORM Nº 54 SHEET Nº _____
PAGE Nº ___ SHEET __ OF __

SUBJECT:

CHART FOR THE SELECTION OF PIPE STRESS FORMS.

COMPUTED BY _____ DATE _____

CHECKED BY _____



SUMMARY SHEET 46
α, β, λ SHEET 48
TABLE FOR SELECTION OF FORMS 49
TABULATION OF DIMENSIONS. 50
SKETCH & OUTLINE COMP. 3-PLANE 51-52
" " " " 1- " 53-54

GROUP NO. _____ FORM NO. 55 SHEET NO. _____

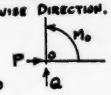
SHEET __ OF __

SUBJECT: SINGLE PLANE PIPE, SIMULTANEOUS EQUATIONS.

COMPUTED BY _____ DATE _____
CHECKED BY _____

① $A_s M_o + A_y P - A_x Q = 0$
② $A_y M_o + A_{y^2} P - A_{xy} Q = EI\Delta x$
③ $A_x M_o + A_{xy} P - A_{x^2} Q = -EI\Delta y$

NOTE: POSITIVE MOMENTS AT "O" IN COUNTER CLOCKWISE DIRECTION.



① + _____ M_o + _____ P − _____ Q = 0
② + _____ M_o + _____ P − _____ Q = + _____
③ + _____ M_o + _____ P − _____ Q = − _____

$\frac{A_y}{A_s}$ = _____ = _____

② + _____ M_o + _____ P − _____ Q = + _____
① Mod. + _____ M_o + _____ P − _____ Q = 0
④ _____ P _____ Q = + _____

$\frac{A_x}{A_s}$ = _____ = _____

③ + _____ M_o + _____ P − _____ Q = − _____
① Mod. + _____ M_o + _____ P − _____ Q = 0
⑤ _____ P _____ Q = − _____

$\frac{\text{COEFF. OF ④ P}}{\text{COEFF. OF ⑤ P}}$ = _____ = _____

④ _____ P _____ Q = + _____
⑤ Mod. _____ P _____ Q = − _____
Q = _____
Q = _____

SUBSTITUTING IN ④ _____
④ _____ P _____ × _____ = _____
_____ P = _____
P = _____

GROUP Nº ___ FORM No 100 SHEET Nº _____
PAGE Nº ___ SHEET ___ OF ___

SUBJECT: PIPE STRESSES FOR SINGLE PLANE PIPE, SIMULTANEOUS EQUATIONS.

PIPE FROM _____ TO _____

CHECKING IN ⑤

⑤ _____ P _____ . Q= _____

⑤ _____ x _____ _____ x _____ = _____

_____ = _____

_____ = _____

NEW OLD

SUBSTITUTING IN ① TO OBTAIN M_o.

① $+A_5M_o+$ _____ P− _____ Q = 0

① $+A_5M_o+$ _____ _____ − _____ _____ = 0

① $+A_5M_o$ _____ _____ = 0

$+A_5M_o$ _____ = 0

+(_____)M_o =

M_o = _____

CHECKING IN ②

② + _____ x _____ + _____ x _____

− _____ x _____ = EIΔx

_____ _____ _____ = EIΔx

NEW EIΔx = _____

OLD EIΔx = _____

DIFFERENCE = _____

CHECKING IN ③

③ + _____ x _____ + _____ x _____

− _____ x _____ = EIΔy

_____ _____ _____ = EIΔy

NEW EIΔy = _____

OLD EIΔy = _____

DIFFERENCE = _____

GROUP NO._____ FORM No 101 SHEET No._____

DATE: _____ COMPUTED BY _____ CHECKED BY _____ SHEET OF____

SUBJECT: DETERMINATION OF FORCES, MOMENTS, AND STRESSES IN A PIPE LYING IN THREE PLANES AND FIXED AT EACH END.

IN PIPE FROM _____ TO _____

① $+\alpha_1 N - \beta_1 P - \alpha_2 Q = EI\Delta_z$
② $-\alpha_2 N - \beta_2 P + \lambda_2 Q = EI\Delta_y$
③ $-\beta_1 N + \beta_3 P - \beta_2 Q = EI\Delta_x$

① + _____ N − _____ P − _____ Q = _____
② − _____ N − _____ P + _____ Q = _____
③ − _____ N + _____ P − _____ Q = _____

$\frac{\alpha_1}{\alpha_2} = \frac{\text{_____}}{\text{_____}} =$ _____

① + _____ N − _____ P − _____ Q = _____
② Mod. − _____ N − _____ P + _____ Q = _____
④ _____ P _____ Q = _____

$\frac{\alpha_1}{\beta_1} = \frac{\text{_____}}{\text{_____}} =$ _____

① + _____ N − _____ P − _____ Q = _____
③ Mod. − _____ N + _____ P − _____ Q = _____
⑤ _____ P _____ Q = _____

GROUP NO. _____ FORM NO. 802 SHEET NO. _____
DATE _____ COMPUTED _____ CHECKED _____ SHEET OF _____

SUBJECT: _____

DETERMINATION OF FORCES IN A THREE PLANE PIPE _____

COMPUTED BY _____ DATE _____

CHECKED BY _____

$$\frac{\text{COEFF. } ④ \text{ OF } P}{\text{COEFF. } ⑤ \text{ OF } P} = \frac{_____}{_____} = _____$$

④ _____ P _____ Q = _____

⑤ MOD _____ P _____ Q = _____

_____ _____ Q = _____

Q = _____

SUBSTITUTING IN ④ TO DETERMINE P

$-(_____ \times Q) = _____$

DIVIDE BY COEFF. OF P IN ④

$_____ = _____$

P = _____

SUBSTITUTING IN ① TO DETERMINE N

$+\beta_1 P = _____ (\beta_1)_K \times _____ (P) = _____$

$+\alpha_2 Q = _____ (\alpha_2)_K _____ (Q) = _____$

$EI\Delta z = _____$ (SUM)

$\alpha_1 N = _____ (\alpha_1)_K _____ N = _____$

N = _____

GROUP Nº ____ FORM Nº 103 A SHEET Nº _____

PAGE Nº ____ SHEET ____ OF ____

SUBJECT Checking of Forces in a Pipe Lying in Three Planes.

COMPUTED BY _____ DATE _____
CHECKED BY _____

CHECKING IN ②

$-\alpha_2 N = -$ _____ × _____ (N) = _____
$-\beta_2 P = -$ _____ × _____ (P) = _____
$+\lambda_2 Q = +$ _____ × _____ (Q) = _____
New $EI\Delta y$ = _____
Former $EI\Delta y$ = _____
Difference = _____

CHECKING IN ③

$-\beta_1 N = -$ _____ × _____ (N) = _____
$+\beta_3 P = +$ _____ × _____ (P) = _____
$-\beta_2 Q = -$ _____ × _____ (Q) = _____
New $EI\Delta x$ = _____
Former $EI\Delta x$ = _____
Difference = _____

GROUP No. _____ FORM No. 104 SHEET No. _____
SHEET _____ OF _____

SUBJECT ... DERIVING THE MOMENTS DEVELOPED IN A THREE PLANE PIPE.

COMPUTED BY ... DATE: ...
CHECKED BY ...

$-B_z\,Q = -$ ____ x ____ (Q) = ____

$+B_y\,N = +$ ____ x ____ (N) = ____

$+B_s\,M_{ox} = +$ ____ x M_{ox} = ____

$M_{ox} =$ ____ LBS IN

$+C_z\,P = +$ ____ x ____ (P) = ____

$-C_x\,N = -$ ____ x ____ (N) = ____

$+C_s\,M_{oy} =$ ____ x M_{oy} = ____

$M_{oy} =$ ____ LBS IN

$-A_y\,P = -$ ____ x ____ (P) = ____

$+A_x\,Q = +$ ____ x ____ (Q) = ____

$+A_s\,M_{oz} =$ ____ x M_{oz} = ____

$M_{oz} =$ ____ LBS IN

$+M_{ox} =$ ____

$+TQ = +$ ____ x ____ (Q) = ____

$-HN = -$ ____ x ____ (N) = ____

$M_{AX} =$ ____ LBS IN

$+M_{oy} =$ ____

$-TP = -$ ____ x ____ (P) = ____

$+LN = +$ ____ x ____ (N) = ____

$M_{AY} =$ ____ LBS IN

$+M_{oz} =$ ____

$+HP = +$ ____ x ____ (P) = ____

$-LQ = -$ ____ x ____ (Q) = ____

$M_{AZ} =$ ____ LBS IN

NOTE: L, H, & T ARE MEASURED ALONG X Y & Z AXES RESPECTIVELY.

GROUP NO. ... FORM NO. 105 SHEET NO. ...

SHEET OF ...

SUBJECT _____

CHECKING FORCES AND MOMENTS IN A THREE PLANE PIPE

COMPUTED BY _____ DATE _____

CHECKED BY _____

$-C_z \cdot Moy$	$= -$	$\times$ ( ) (Moy)	$=$
$+Ay\ Moz$	$= +$	$\times$ ( ) (Moz)	$=$
$+(Ay^2 + C_z^2)P$	$= +$	$\times$ ( ) (P)	$=$
$-Axy\ Q$	$= -$	$\times$ ( ) (Q)	$=$
$-Cx_z\ N$	$= -$	$\times$ ( ) (N)	$=$
		NEW $EI\Delta x$	$=$
		FORMER $EI\Delta x$	$=$
		DIFFERENCE	$=$
$+B_z Mox$	$= +$	$\times$ ( ) (Mox)	$=$
$-Ax\ Moz$	$= -$	$\times$ ( ) (Moz)	$=$
$-Axy\ P$	$= -$	$\times$ ( ) (P)	$=$
$+(Ax^2 + B_z^2)Q$	$= +$	$\times$ ( ) (Q)	$=$
$-By_z\ N$	$= -$	$\times$ ( ) (N)	$=$
		NEW $EI\Delta y$	$=$
		FORMER $EI\Delta y$	$=$
		DIFFERENCE	$=$
$-By\ Mox$	$= -$	$\times$ ( ) (Mox)	$=$
$+Cx\ Moy$	$= +$	$\times$ ( ) (Moy)	$=$
$-Cx_z P$	$= -$	$\times$ ( ) (P)	$=$
$-By_z Q$	$= -$	$\times$ ( ) (Q)	$=$
$+(Cx^2 + By^2)N$	$= +$	$\times$ ( ) (N)	$=$
		NEW $EI\Delta z$	$=$
		FORMER $EI\Delta z$	$=$
		DIFFERENCE	$=$

GROUP Nº _____ FORM No. 106 SHEET Nº _____

SHEET __ OF __

SUBJECT FORCES - MOMENTS - STRESSES IN PIPE (SINGLE PLANE)

COMPUTED BY _____ DATE _____
CHECKED BY _____

Δx = _____ Δy = _____
= _____ = _____
= _____ = _____

$\Delta\psi$ = _____

P = _____
= _____ = P = _____

Q = _____
= _____ = Q = _____

M_o = _____
= _____ = M_o = _____

POINT OR SECT	H	L	M_o	+HP	−LQ	MOMENT AT POINT	BENDING STRESS	LONG. PRESS STRESS	TOTAL LONG. STRESS
O									
A									

GROUP Nº _____ FORM Nº 112 SHEET Nº _____
SHEET _____ OF _____

"-PIPE

SUBJECT DETERMINATION OF STRESSES FOR A PIPE IN THREE PLANES

COMPUTED BY _____ DATE _____
CHECKED BY _____

"O" END AT	"A" END AT
BENDING M_o	M_A
$(M_o)^2$ +	$(M_A)^2$ +
BENDING M_o	M_A
$(M_o)^2$ +	$(M_A)^2$ +
ΣM_o	ΣM_A
$M_o = \sqrt{\Sigma M_o}$	$M_A = \sqrt{\Sigma M_A}$
r_i/I	r_i/I
LONG. STRESS DUE TO BENDING $\frac{r_i\sqrt{\Sigma}}{I}$ +	$\frac{r_i\sqrt{\Sigma}}{I}$ +
P	P
$r_i/2t$	$r_i/2t$
LONG. STRESS DUE TO PRESS. $\frac{Pr_i}{2t}$ +	$\frac{Pr_i}{2t}$ +
σ_o = TOTAL LONG. STRESS	σ_A
TORSION M_o	M_A
$r_o/2I$	$r_o/2I$
T_o = SHEARING STRESS $\frac{r_o M_o}{2I}$	$\frac{r_o M_A}{2I} = T_A$
P	P
r_i/t	r_i/t
σ_t = HOOP STRESS $\frac{Pr_i}{t}$	$\frac{Pr_i}{t} = \sigma_t$
σ_o	σ_A
σ_o^2 +	σ_A^2 +
σ_t	σ_t
σ_t^2 +	σ_t^2 +
T_o	T_A
T_o^2	T_A^2
$3T_o^2$ +	$3T_A^2$ +
$\sigma_o\sigma_t$ −	$\sigma_A\sigma_t$ −
$\Sigma(\sigma+T)$	$\Sigma(\sigma+T)$
$\sigma_{eq.}$ = EQUIVALENT STRESS = $\sqrt{\Sigma}$	$\sqrt{\Sigma(\sigma+T)} = \sigma_{eq.}$

GROUP NO. ____ FORM NO 113 SHEET NO. ____
PAGE NO. SHEET ___ OF ___

___" PIPE ___

SUBJECT: DETERMINATION OF STRESSES FOR A PIPE IN THREE PLANES DUE TO CHANGE IN TEMPERATURE

COMPUTED BY: _____ DATE: _____
CHECKED BY: _____

ORIGINAL TEMPERATURE _____°F. NEW TEMPERATURE _____°F.
ORIGINAL $E = E_o =$ _____ NEW $E = E_N =$ _____
ORIGINAL EXPANSION $= \delta_o =$ _____ NEW EXPANSION $= \delta_N =$ _____

$$\text{FACTOR} = \frac{E_N \times \delta_N}{E_o \times \delta_o} = \frac{\quad}{\quad} \times \frac{\quad}{\quad} = \quad$$

"O" END AT _____	"A" END AT _____
BENDING STRESS _____	BENDING STRESS _____
FACTOR $\times$ _____	FACTOR $\times$ _____
LONG. STRESS DUE TO BENDING + _____	LONG. STRESS DUE TO BENDING + _____
P _____	P _____
$r/2t \times$ _____	$r/2t \times$ _____
LONG. STRESS DUE TO PRESS. $= \frac{Pr}{2t}$ + _____	LONG. STRESS DUE TO PRESS. $= \frac{Pr}{2t}$ + _____
σ_o = TOTAL LONG. STRESS _____	σ_A = TOTAL LONG. STRESS _____
TORSION STRESS _____	TORSION STRESS _____
FACTOR $\times$ _____	FACTOR $\times$ _____
τ_o = SHEAR STRESS DUE TO TORSION _____	τ_A = SHEAR STRESS DUE TO TORSION _____
P _____	P _____
$3/4 \times$ _____	$3/4 \times$ _____
σ_t = HOOP STRESS $= \frac{Pr}{t}$ _____	σ_t = HOOP STRESS $= \frac{Pr}{t}$ _____
σ_o _____	σ_A _____
σ_o^2 + _____	σ_A^2 + _____
σ_t _____	σ_t _____
σ_t^2 + _____	σ_t^2 + _____
τ_o _____	τ_A _____
τ_o^2 _____	τ_A^2 _____
$3\tau_o^2$ + _____	$3\tau_A^2$ + _____
$\sigma_o\sigma_t$ − _____	$\sigma_A\sigma_t$ − _____
$\Sigma(\sigma+\tau) =$ _____	$\Sigma(\sigma+\tau) =$ _____
$\sigma_{eq} = \sqrt{\Sigma(\sigma+\tau)} =$ _____	$\sigma_{eq} = \sqrt{\Sigma(\sigma+\tau)} =$ _____
EQUIVALENT STRESS = _____	EQUIVALENT STRESS = _____

GROUP NO. _____ FORM NO. 114 SHEET NO. _____
PAGE NO. SHEET OF ____

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THE WEIGHT FACTOR IN THE DESIGN OF RADIO EQUIPMENT FOR NAVAL AIRCRAFT.

PART II

By Wm. Bruce Wells.*

Due to its length, the Part II of this paper has had to be divided into Parts II and III. Part I was published in the November, 1940, issue of the Journal. *Part III will be published in a later issue.*

Before examining in greater detail the factors contributing to weight in aircraft radio equipment and the various methods employed toward weight reduction in the apparatus itself, it may be advisable to review the antenna situation in naval aircraft together with a brief further discussion of the reasons underlying the present trend, since reduction of antenna facilities so largely militates against appreciable decreases in the size and weight of radio equipment when low or intermediate radio frequencies must be covered.

We have already seen, in the introductory part of the article (November, 1940), how the physical and electrical properties of shortened antennas can seriously affect the performance and bulk of the radio equipment, especially the radio transmitter, in the effort to reduce aerodynamic drag and conform to the many other requirements peculiar to the Naval service. Thus, the exactions of modern Naval aircraft have largely curtailed the electrical effectiveness of the antenna with the result that the bulk and weight of the equipment inside the airplane has tended to increase. This reduced size and effectiveness of antennas has been occasioned chiefly by aerodynamic considerations. In order to diminish parasite drag, not only the size but the form and

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height have become electrically less favorable. Thus, the V and W-shaped antennas have been replaced by straight fore-and-aft antennas whose distance from the metal fuselage is still further reduced by the necessity of short supporting masts. As mentioned earlier, the fore-and-aft antenna has been additionally reduced in length by location of the supporting mast well aft of the leading edge of the wing where it offers less drag. The trailing wire antenna has already been disposed of as undesirable owing to its drag at high speed, etc., together with the difficulty of reeling it in. Electrically-driven winches have been devised for this purpose but these necessarily add to the weight that must be carried. However, in order to provide reliable long-distance communication on low or intermediate frequencies (300-1500 Kcs.) and to provide signals suitable for accurate radio D. F. bearings when occasion demands, the weighted trailing wire must be retained, since very little power can be delivered into a fixed antenna at these frequencies, particularly at high altitudes. In modern patrol planes the trailing wire also involves additional deadweight in the form of long, heavily insulated fairlead structures. The exposed direction-finder loop antenna employed in large aircraft of the patrol type has also undergone a metamorphosis by enclosure in a streamlined housing in the interests of reduced drag, loop vibration, and greater azimuthal training stability, with an attendant improvement in accuracy of bearings on the part of the operator at present increased speeds. By enclosing the loop in a nacelle, the electrical performance is slightly reduced at the lower frequencies as a result of decreased loop diameter but this is somewhat compensated for by the physical ability to obtain finer azimuth adjustments. Relocation of the loop structure to further decrease drag and to provide a more favorable electrical position on the airplane would require remote control with the additional weight represented by the control mechanisms.

Thus, to satisfy present aircraft demands for higher speeds, enhanced maneuverability, sustained operation at higher altitudes, greater range and endurance, all with improved radio communication and navigational facilities, and with increased loads comprising bombs, armament and ammunition, armor, fuel tank protection, supercharging and oxygen equipment, etc., the eternal conflict between the aircraft designer and radio equipment designer

goes on, while both must face the multi-pronged horns of an almost uncompromising dilemma. This brings us to the question of Drag *vs.* Weight in an airplane, and a few words concerning drag and weight considerations in aircraft and their relation to the radio problem may be of interest.

While the aircraft designer must choose between the conflicting requirements of minimum drag and minimum weight so also is the radio engineer confronted with the problem of trying to build increased performance to a reduced size and weight. It is fortunate for the radio designer that, insofar as contemporary aircraft designs are concerned, it is only when the contemplated weight, based on the weights of existing radio gear, becomes exceeded that the designed airplane performance is compromised. However, for new aircraft designs wherein the demand for improved aerodynamic (and radio!) performance is increasing, the weight of the radio installation must be reduced below existing limits. Even in present and past designs radio weight must give way to allow the increases of other useful load in the form of guns, oxygen and pressurizing equipment for high altitude flying, ammunition, etc. In short, for future radio equipments more emphasis must be placed on size and weight reduction and less on engineering refinements in an attempt to attain electrical perfection.

Aerodynamically speaking, almost any reduction in drag justifies the necessary extra weight. However, with a limit to landing speed and take-off speed such as may be imposed by aircraft carrier, cruiser and battleship operation, extra structural weight results either in reduced useful load or greater requisite wing area, and some of the advantage of drag reduction is lost where increased weight is required thereby. Broadly speaking, in practically every external detail, the aircraft designer must choose between a light, high drag design and a heavier design with less drag. In somewhat similar fashion, the effort to improve the performance of radio equipment and yet reduce its weight resolves itself into a series of seemingly unending compromises.

Accordingly, as the size of the radio antennas is diminished, the weight of associated equipment rises due to a multiplicity of causes. As may be observed in Figures 7 and 8 of the first part of this article, one of which is reproduced in part here,

(Figure 1) smaller antenna dimensions reduce the antenna efficiency and radiated power output. This necessitates at least a corresponding increase of power available from the radio transmitter which then becomes larger and heavier and also places a greater drain on the engine power plant which in turn increases the fuel load necessary, and so on. Somewhat similarly, in the special case of exposed D. F. loop antennas, an airfoil housing

FIXED ANTENNA

3000-10,000 KC.

A = 20' 4" FLAT TOP "T" TYPE RESONANT FREQ. 11.0 MCS.
B = 14' 6" PER LEG "V" TYPE RESONANT FREQ. 7.7 MCS.

FREQUENCY - KCS.	3000	3000	5000	5000	6000	6000	9000	9000
LENGTH - FEET	A	B	A	B	A	B	A	B
C ANT. $\mu\mu$ fd.	70	116	80	166	87	220	130	
R ANT. - OHMS.	1.7	1.26	2.2	1.4	2.9	1.7	12	3
L ANT. LOAD - μ HEN.	40	24.5	13	6.2	8	3.2	2.5	CAP.
X_{L_a} - OHMS	752	460	406	195	300	121	141	
R_{L_a} - OHMS	5	3	2.7	1.3	2	0.8	0.94	
$R_a + R_{L_a}$	6.7	4.26	4.9	2.7	4.9	2.5	12.9	
I_a FOR 55 WATT TUBE C W OUTPUT	2.86	3.59	3.35	4.5	3.35	4.7	2.06	
$I_a^2 R_a$ ANT. WATTS	14	16.2	24.6	28	32.5	37.5	51	
$I_a R_{L_a}$ ANT. LOSS - WATTS	41	39	30.4	27	22.5	17.5	4	
$I_a X_{L_a}$ R M S VOLTS C W	2160	1650	1360	878	1000			
$I_a X_{L_a}$ — PEAK R.F. VOLTS PHONE AT PEAK OF 75% MODULATION, 35 WATT CARRIER.	4280	3270	2700	1740	2000	1120	572	

FIGURE 1.

or blister enclosing the loop must be resorted to in order to reduce parasite drag. (See Figure 2.)

This procedure reduces parasite drag but generally requires that the diameter and area of the shielded loop itself be reduced with an accompanying reduction in efficiency, since the resultant voltage acting around an untuned loop is equal to

$$2\pi\epsilon N \frac{\text{(loop area)}}{\lambda} \cdot \cos\theta, \text{ where } \epsilon = \text{strength}$$

of the radio wave in volts per meter, N = number of turns in the loop, λ = wave length of radio wave in meters, and θ = direction of travel of wave with respect to plane of loop. The effectiveness of the loop as a means of abstracting energy from radio waves is its "effective height," which is the above resultant loop voltage divided by the field strength ϵ. It is evident that for a given frequency or wave length, and direction of wave travel, the effective height decreases as the square of the decrease in diameter. This can be partly compensated for by increasing the number of turns, N, but chiefly by improvement in the "Q" of the loop and in the electrical coupling to the radio receiver. The "Q", which is approximately the reciprocal of the coil power factor, is expressed as the ratio of coil reactance to coil resistance, $\omega L/R$, where ωL is the coil reactance and R is the effective coil resistance and includes dielectric loss and losses due to skin effect. With tuned loop circuits, as are generally employed, the induced voltage in the loop coil is increased by the effective "Q" of the circuit when tuned to resonance at the frequency of the incoming signal, and the voltage across the coil becomes Q times the induced voltage. Since the effectiveness of an aural (or visual) null type direction finder depends upon the ability to obtain a reasonably sharp, well-defined signal minimum which can be readily "split", and as this in turn depends upon the signal-to-noise ratio at the receiver output, it will be recognized that any effort to increase the signal sensitivity and improve the signal/noise ratio becomes especially important with loops of small size.

The total weight of such equipment is increased not only by the additional weight of the streamline enclosure but also by the electrical means made necessary to maintain satisfactory radio direction-finding performance. These generally take the form of methods for reducing the radiofrequency resistance with consequent improvement of the "Q" of the loop and associated circuit, such as by using a powdered iron core in the loop, increased number of turns in the loop winding and improved coil form factor, lower loss insulation, improved coupling means using iron dust cored transformers and more efficient matching circuits, etc. Since size is fixed, any attempt to increase Q usually means added weight. For example, a suitable ceramic insulation may be substituted for the higher loss but lighter weight phenolic insulation,

larger conductors and/or more strands therein may be employed to lower the R. F. coil resistance, while powdered iron coil cores for increasing the effective inductance also add their share to the total weight. If the loop inductance is permitted to decrease, the variable tuning capacitor must be made larger, and heavier, in order to insure resonance at the lower radio frequencies. If, for aerodynamic reasons, the loop must be located in a position inconvenient or inaccessible to the radioman or navigator, the increase in weight comes about by the necessity for remote control mechanisms (either mechanical, hydraulic or electric), including electric repeater azimuth indicators, and increased radio frequency transmission line and cable lengths. A typical remotely controlled, exposed loop installation may weigh as much as 25 or 30 pounds more than its local control equivalent, which will have a total weight in the neighborhood of 50 pounds, including radio receiver. The increase in weight of a typical direction finder installation having manual, local control and the loop enclosed in an external streamline housing is roughly 7 pounds over that for a similar installation having an exposed circular loop of approximately 14 inches outer diameter, including its metallic shield with circular cross-section. The added weight represents an increase of about 9 per cent for the particular D. F. installation involved. On the other hand, the aerodynamic drag of the loop nacelle employed with the former installation is only about 7 pounds at 250 miles-per-hour, whereas, although exact data are not available to the writer, the drag due to the exposed loop would considerably exceed this value with the plane of the loop parallel to the wind stream, and would be increased by four or five times when the plane of the loop is in the athwartships position. It should be noted that the latter would be the normal position of the loop when used for homing. The foregoing values may be increased somewhat by interference drag in many installations. The parasite drag at other speeds will vary directly as V^2, the square of the velocity. By way of comparison, the drag of the average V type antenna installation including masts, insulators, lead-in wires, etc., on an airplane having about a thirty-five foot span would be around 65 pounds at 250 m.p.h. Figure 3 shows a typical exposed loop D. F. with interior rotating shaft for direct, manual control.

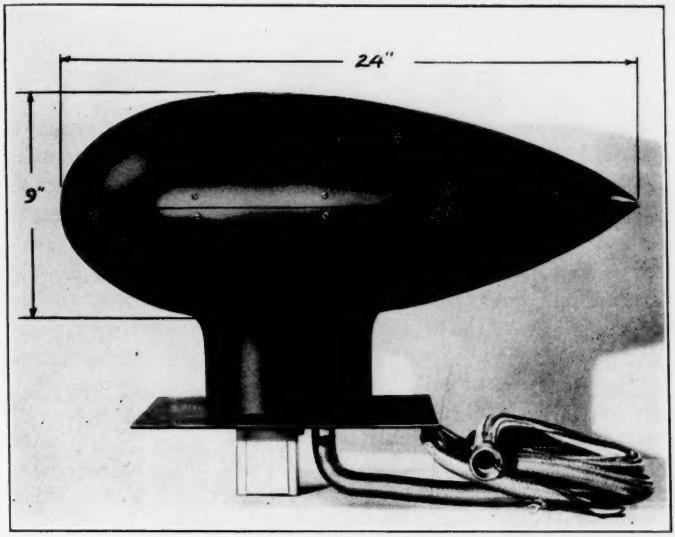


FIGURE 2.

Streamlined D. F. loop nacelle of a recent commercial design.

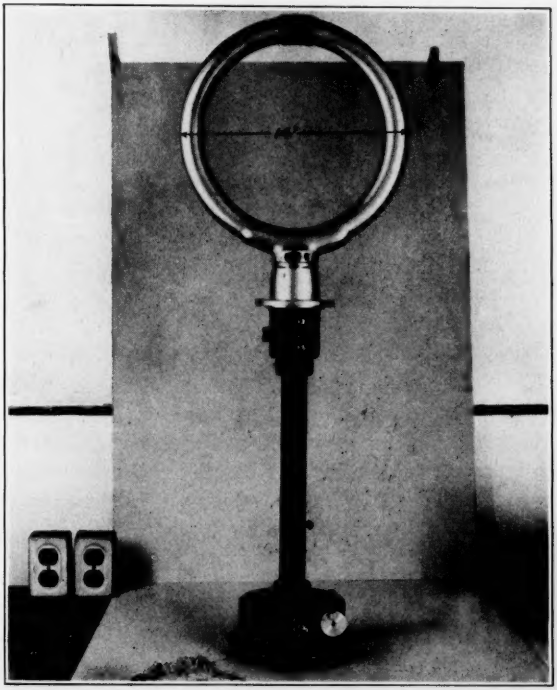


FIGURE 3.

A Typical Exposed D. F. Loop Structure with its Rotating Shaft.

Although parasite drag becomes of greatest importance at the maximum speed of an airplane, relatively small changes in weight, if they do not directly affect the parasite drag, have a minor influence on the maximum velocity, as illustrated in Figure 4, which is representative of an airplane's behavior as influenced by weight. While the induced drag of the wings is proportional to the span loading, it varies inversely as V^2. The intersection of the available thrust horsepower curve and the curve of thrust horsepower required for a given gross weight, represents the maximum

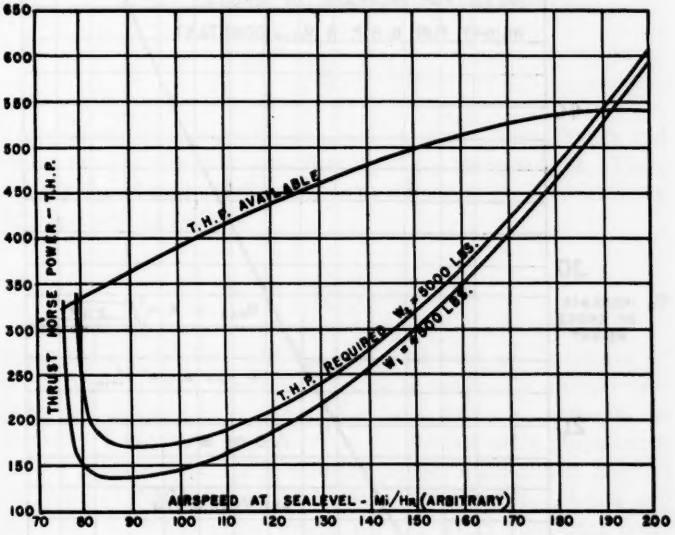


FIGURE 4.

speed for the particular conditions involved. The difference between the available T.H.P. and required T.H.P. is the excess power available for climb, and this, divided by the gross weight gives the rate of climb at sea-level or "initial rate of climb." The maximum speed is related to the parasite drag, for a specified set of conditions by the following expression,

$$V_{M2} = \sqrt[3]{\frac{D_{p1}}{D_{p2}}} \times V_{M1},$$

from which the importance of drag reduction toward increasing the maximum velocity of the airplane may be readily appreciated. The maximum velocities are inversely proportional to the cube root of the drags. If the weight can be decreased, such as by structural refinements, lighter materials, etc., the resultant airplane will be smaller and have less drag. Figure 5 shows a very

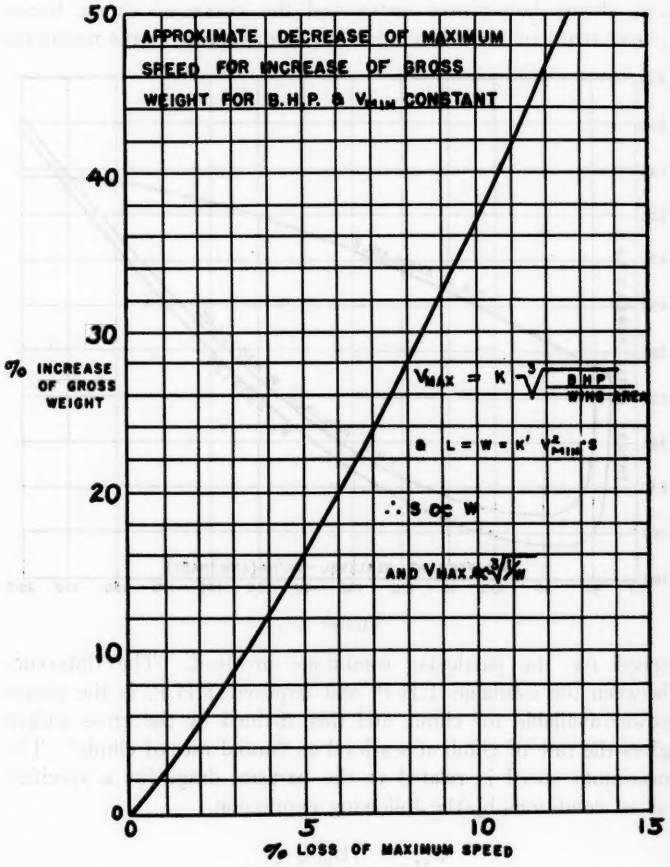


FIGURE 5.

approximate curve of the decrease in maximum speed with increased gross weight. This assumes that V_{Max} is proportional to the cube root of the ratio of B.H.P. to Wing Area and that for a constant speed the Lift equals the Weight and is proportional to the product of the Wing Area and the square of the minimum or stalling velocity.

For a given wing area, equivalent wing aspect ratio, thrust horsepower, propeller efficiency, and air density, the parasite drag has practically no effect on the stalling speed; however, weight affects the stalling speed in accordance with the relation

$$V_{S2} = \sqrt{\frac{W_2}{W_1}} \times V_{S1}.$$

The take-off distance and speed, rate of climb, ceiling, range and endurance are similarly dependent upon the gross weight. Thus, the relation of take-off distance and a change in gross weight may be expressed as follows,

$$S_1 = k \cdot \left(\frac{W_1}{W_2}\right)^2 \times S_2,$$

while the ground run with no wind will vary as the square of the take-off speed. The rate of climb at sealevel is directly proportional to the excess horsepower (the difference between the available T.H.P. and required T.H.P.) but inversely proportional to the gross weight. The bombing range is dependent only on the ratio of the load released to the initial gross load, hence any reduction in initial gross weight permits either additional bomb load or greater bombing range. The weight load may become of considerable concern in multi-engine flying boats of the patrol-bomber class when one or more engines fail. For further information on Aerodynamics the reader is referred to the excellent text on this subject—"Engineering Aerodynamics"—by Commander W. S. Diehl, U. S. N. (Ronald Press Co., N. Y.)

A few examples may better serve to illustrate the importance of drag and weight reduction in an airplane. A decrease in weight per horsepower in an airplane will generally mean increased ground speed, increased rate of climb, decreased time of climb to

a given altitude, and a raising of the service ceiling. At the maximum speed of a certain airplane, 10 per cent additional weight would be justified if the parasite and minimum profile drag could thereby be reduced 1 per cent; at the cruising speed, a 1 per cent reduction in the minimum drag would justify a 5 per cent increase in gross weight. For a given "pay" load and initial gross load, including fuel, the cruising range might be increased by 50 per cent, at the same cruising speed obtainable with the same horsepower, if the minimum drag were reduced by 15 per cent, thus permitting an increase in gross weight and therefore fuel load after subtracting a reasonable amount for added structural weight, fuel tanks and power plant. However, with limited loading and take-off speeds a true evaluation of the relative importance of weight and drag is not possible without many assumptions regarding wing area, use of high-lift devices, etc., and furthermore, it must be remembered that when the drag is decreased, additional weight is aerodynamically justified at any given speed (above stalling) only if the parasite and profile drag is at least partially reduced thereby. The Lift/Drag ratio varies with the load, amongst other things, so that any device which directly increases the head resistance of an airplane might be considered as an equivalent weight equal to the weight increase caused by the device multiplied by L/D, and the sum of this equivalent and the actual weight is the effective weight.

Actual practice in recent years has been to improve airplane speed through increased wing loading, together with the use of high-life devices to restrain the stalling speed. It has been stated, after careful analysis, that further improvements in speed will be accomplished chiefly by elimination of wasted and unnecessary parasite drag and by high-altitude flying with supercharged engines and cabins; the latter contributing over twice as much toward speed increase as the former. The contribution of reduced drag toward higher speed, on the other hand, would be two to three times that possible by weight reduction. That last increment of speed, however small, may well become the deciding factor in aerial combat.

The electrical power drain required by the radio equipment might be considered as detracting from the available thrust-horsepower or as increasing the power loading, Wg/BHP. How-

ever, the electrical load required by the usual aircraft radio installation is so small in comparison with the other electrical and mechanical loads on the engine(s) and the available horsepower that it may safely be neglected, except possibly insofar as additional weight may be required for emergency operation after main engine failure, forced landing, available power when at anchor, etc. For modern fighter airplanes the L/D ranges from 10 or 12 or larger at 60 per cent maximum speed to as high as 18 for airplanes of the patrol bomber class. In general, at a constant velocity, the ratio of effective weight to actual weight of modern aircraft electrical power supply systems for radio is quite small, normally less than 2:1. For a given set of conditions, diversion of power from the propeller, i.e. use of the main engine is generally preferable to supplying the power from a separate, auxiliary engine power source insofar as better airplane performance in regard to stalling speed, rate of climb, ceiling, and take-off is concerned, but the latter practice permits a larger relative value of high speed.

The airplane's flight performance, including the employment of its ordnance, is necessarily the paramount consideration in aircraft operation; however, radio performance, which includes the effectiveness of communications, aids to air navigation, instrument landing, etc., is tactically indispensabe to the successful conduct of naval operations and organization.

Next to the antenna limitations necessitated by aerodynamic considerations, the demands of the naval service imposed by its operating requirements and circumstances are otherwise the chief contributors to weight in the radio equipment, and these requirements constitute the principal handicap in any effort to achieve further size and weight reductions of substantial magnitude in such equipment.

Naval Service Requirements.

An investigation of service requirements and service conditions is therefore first essential to a proper understanding of their effect upon the existing weight of aircraft radio equipment. In the order of their relative importance as regards radio weight, they are, assuming minimum required radiated fields and received field intensities:

a. Employment of low frequencies (below approx. 3 Megacycles).
b. Frequency range to be included.
c. Altitude requirements, for given aircraft transmitter power output.
d. Remote control of apparatus.
e. Number of simultaneous operating channels necessary.
f. Frequency stability and tunability of transmitters.
g. Selectivity, tunability, and stability of receivers.
h. Vibration and acceleration.
i. Reliability.
j. Full-automatic control.
k. Space available for installation of equipment.
l. Maximum working distance.

It might appear that the last should be first as a weight determining influence, but when the minimum transmitter power output necessary to ensure satisfactory reception is established (usually on the basis of past operating experience under service conditions), this requirement is reasonably satisfied since the field strength at a distance from the transmitting antenna is proportional to the square-root of the *radiated* power. In this connection, it is noteworthy that, for a given set of conditions, and assuming receiver input levels sufficiently high to produce linearity in the receiver, if the radiated power is doubled, the receiver output is increased by 3 decibels, an increment which would just be barely perceptible under usual aircraft operating conditions of high ambient acoustic noise levels and of physical fatigue. Also, despite the present demand for increased flying range, the electrical limitations on transmitter output power are predetermined by the antenna size, frequencies to be used, and maximum altitude requirements. The size and shape of the space or spaces available in the airplane are ordinarily fixed quantities for each type of craft and therefore determine to a great extent the density, or weight per unit volume, of the equipment, as distinct from the total weight of the complete installation. Incidentally, the shape or form of the space available for radio equipment influences the weight of the apparatus somewhat since the size and disposition of the components within a unit may be affected. Thus, for a lo-

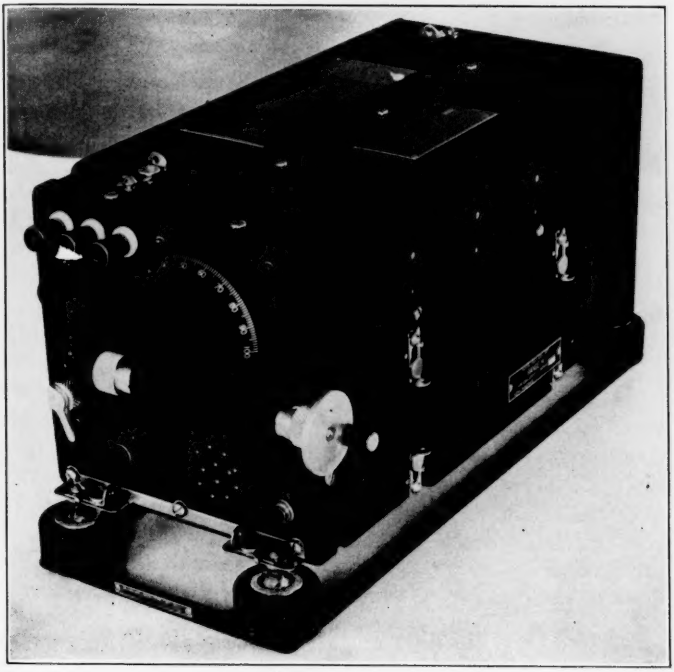


FIGURE 6.

A Typical Aircraft Radio Receiver of the T.R.F., Plug-in Coil Type. Density, as shown = 0.016 pounds per cubic inch.

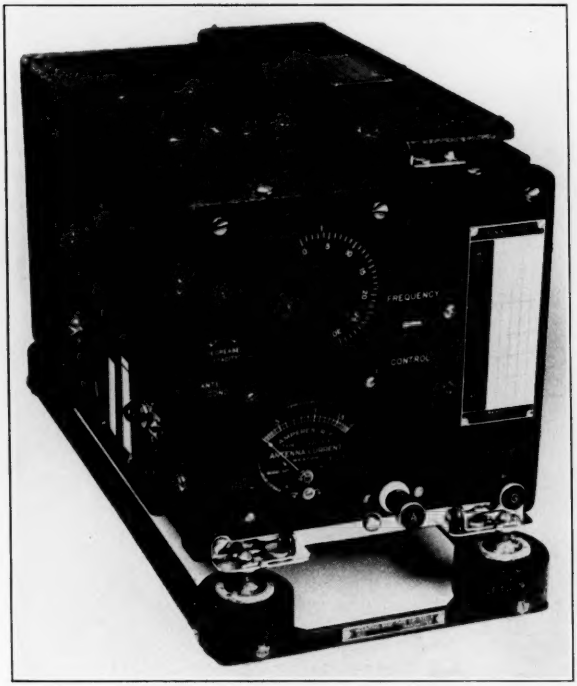


FIGURE 7.

Low Power Transmitter, employing Plug-in Inductance Coils. Density, as shown = 0.017 pounds per cubic inch.

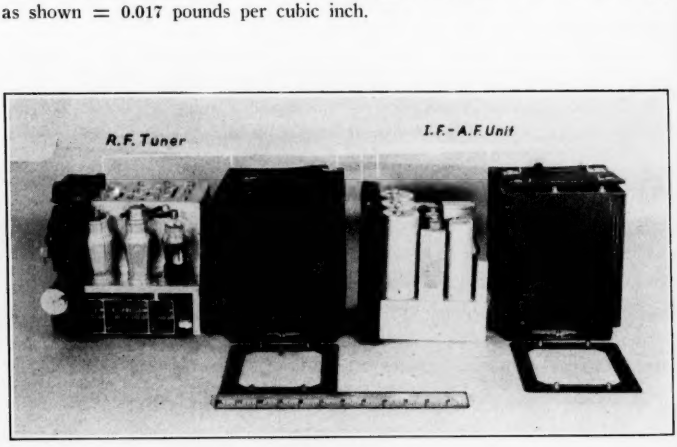


FIGURE 8.

A Double-Unit Superheterodyne Receiver.

cally controlled equipment the control panel area becomes highly important and as this is reduced, or vice versa, the internal parts must be displaced in the direction either of the depth or the width dimension, thereby often increasing the length and size of control shafts, cables, insulation, number of insulators, amount and degree of internal electrical shielding, length of wiring, etc. The form factor adopted for the smaller Naval aircraft radio equipments (receivers and low-power transmitters) has been such as to conform with the major and minor axes of the airplane fuselage, with the front or operating face of the equipment facing either fore or aft as installed. Hence, the length of the unit lies along the major fore-and-aft dimension of the airplane and the longest dimension of such equipment is customarily the depth, with the width restricted as necessary, depending upon the maximum fuselage width and contour and amount of equipment installed. The height is generally set by other considerations which will be discussed later. This form (see Figures 6 and 7) permits either local control of the equipment or remote control with cables and linkages coming out of either the front or rear panel of the unit. Fortunately, this arrangement is also electrically more favorable and convenient in many cases than that of a wider, shallow-depth form, although for greatest tunability and number of manual controls a relatively large panel area becomes necessary. For medium and high-power transmitters (nominal C. W. output 100 watts and above), which usually have been locally controlled, panel area is important and either width or height becomes the major dimension. The width of medium-power transmitters for use in small aircraft is usually the major dimension and such equipment is mounted athwartships with the operating panel facing aft.

In many of the later installations, size and bulk are of greater importance than weight, and where space is thus restricted, additional weight becomes necessary in the form of multiple shock mountings, interconnecting cables and attachments, and duplication of certain parts because the equipment must then be separated into several units, perhaps smaller than otherwise, but which cannot be mounted adjacently. Oddly enough, this is becoming increasingly true of larger Naval aircraft such as scouting and bombing planes and flying boats, while in the case of fighter

planes weight assumes the greater significance. As will be explained later on, size bears a fairly definite relation to the weight of any unit of equipment. Figures 6 and 7 represent excellent examples of small size and weight with moderate compactness per unit, but having limited power and a limited frequency range depending upon the number of separate, plug-in coil sets carried. Figure 8 shows a useful type of arrangement which facilitates installation and overcomes many installation difficulties. In this equipment, a superheterodyne type of multi-band receiver covering a wide frequency range, the R. F. tuning section may be mounted separately from the intermediate-frequency and audio frequency amplifier unit.

The need for radio aids to air navigation, instrument landing, radio-altimeters, etc., for certain missions and classes of aircraft obviously adds directly to the gross weight of radio gear on board, although some functions may be combined with the duties of the basic apparatus. The amount of added weight will depend upon the number and degree of such functions deemed requisite in the airplane.

As a weight factor, operation in variable and high humidities and in the presence of salt spray adds comparatively little, and in any event, precautions to minimize their effects would be observed as a matter of good engineering practice. These include the filling and sealing of transformers, fixed capacitors, etc., in suitable non-hygroscopic compounds, impregnation of R. F. coils and chokes, employment of corrosion resistant metal alloys or suitable platings, impregnated or inherently non-hygroscopic insulation, moisture-proof wiring, the positioning of critical components and a type of construction such that collection of moisture is minimized, and so on.

Low Frequencies, Altitude, and Frequency Range.

Chiefest offender toward weight contribution is the provision of low frequencies in the equipment, particularly in the transmitter, and for a fixed antenna size and specified power output, the need for low frequency operation constitutes the most severe requirement from a design viewpoint. This requirement is made the more severe by present high altitude flying rquirements (30,000 feet and above) but this again is a circumstance beyond the

control of the radio engineer. As has been mentioned previously, the use of low frequencies results in increased inductive loading in the transmitter antenna circuit which then requires increased size of equipment and heavier antenna circuit components —loading coil, capacitors, insulation, etc. Since the antennas on modern Naval aircraft are limited and consequently the useful radiation from the radio transmitters, the loading inductance losses consisting of coil heat ($I_a^2R_L$) losses, leakage, dielectric losses, corona, etc., comprise the major portion of the transmitter output power at low frequencies. The loss resistance in the antenna circuit must be considered as being in series with the radiation resistance of the antenna proper, the latter becoming quite small at low frequencies, as may be seen from the curves presented in the first part of this article, and at resonance the antenna current is accordingly limited thereby. Also, the antenna efficiency as a radiator is expressed by the ratio $\frac{R_r}{R_r + R_L}$ where R_r is the antenna radiation resistance and R_L is the loss resistance of the circuit, and this in turn varies with the ratio of height or effective vertical length to the wavelength, all of which means that the loading losses must be minimized if any useful amount of power is to be radiated at low frequencies. Increasing the output power capability of the transmitter alone would increase the radiated power but at the same or less efficiency, and there is consequently a practical limit to this method of approach. In order to avoid exorbitant loading losses an improvement of the "Q" of the antenna loading must be made, with attendant increase of loading coil and/or tuning capacitor sizes (and their weights) which are already of substantial proportions as required by the lowest operating frequency to be employed. Since a higher R. F. voltage will be developed at the antenna end of this loading coil, the spacing between the coil and adjacent parts, separation of conductors, and clearances to the metal shielding or case enclosing the equipment, must be enlarged. These requirements apply equally to the endeavor to maintain as large a value of coil Q as possible since eddy current and induction losses as well as corona and arc-over must be minimized. It is well to remember, that coil efficiency, or Q, is directly proportional to

the dimensions and shape of the coil, the extent and spread of its induction field, and the amount and proximity of metal and loss-producing material within this field. The weight of the equipment as a whole becomes proportionately greater as its total volume is increased, for several reasons. For example, the cabinet dimensions, frame size, amount of bracing, and gauge of metal must be increased; more rivets, welding, etc., added; larger and heavier insulation provided, et al. The use of smaller coils having a comparatively low Q may result in a more compact transmitter but at the expense of reduced overall efficiency, thereby necessitating more vacuum tubes or larger tubes having higher power rating, etc., and hence the density (weight per unit volume) of the equipment will be greater. (See Figure 9.) Present requirements of higher ceilings for naval aircraft add further to the space demands owing to the reduced dielectric strength of air with increase of altitude.

For a fixed maximum vacuum tube power output from the final amplifier of the transmitter, as is normally the case, a larger Q for the antenna circuit loading inductance results in increased antenna current, which, combined with the large inductive reactance necessary to resonate a small antenna at low frequencies, act to raise the radio-frequency voltage (and also tend to aggravate somewhat the surface leakage and dielectric losses in the antenna circuit), since the R. F. voltage is equal to the product of the two. Referring to Figure 1, it may be seen that at the lowest frequencies recorded, for a designated tube output power, the power loss in the loading inductance is nearly three times the power radiated from the fore-and-aft antenna, while the peak R. F. potential ($\sqrt{2}$ times effective value) at the antenna end of the load coil is 3060 volts for the CW telegraph condition, with a maximum peak voltage on telephone of 4280 volts! In this particular case the loading coil Q is 150, $\left(\frac{752}{5}\right)$ and considered quite good for the type of equipment involved, which is representative of medium power transmitters suitable for small scouting planes. At still lower frequencies the difficulties would be greatly aggravated because the inductance, and inductive reactance ($2\pi fL$), necessary to maintain antenna circuit resonance is in-

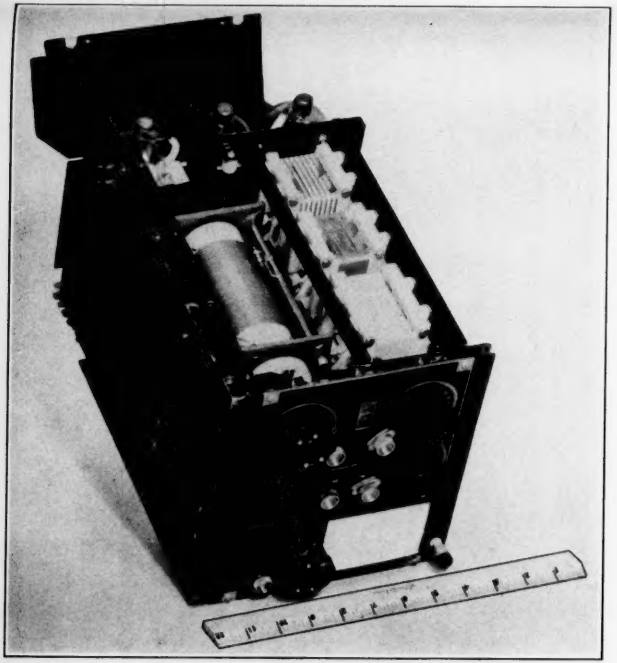


FIGURE 9.

H. F. Section of Aircraft Transmitter of an Obsolescent Type.
Density = 0.020 pounds per cubic inch.

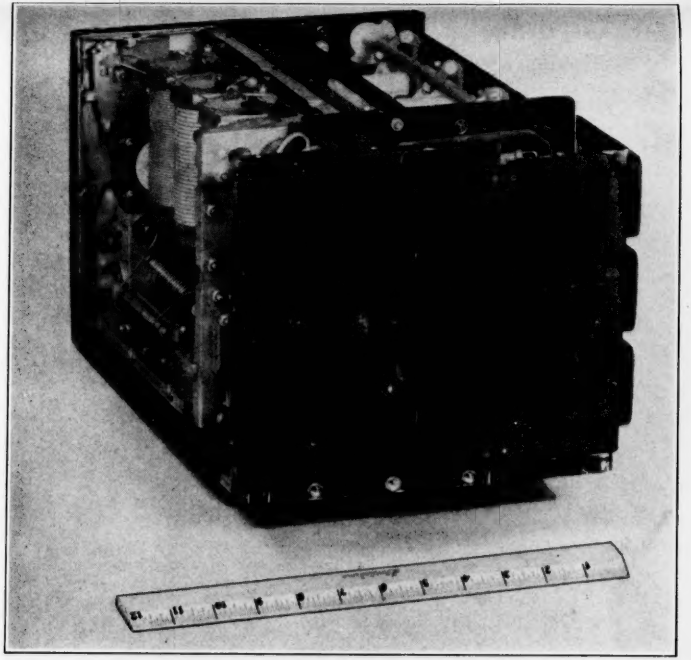


FIGURE 9A.

Rear view, under side of H. F. unit as shown in Figure 8.

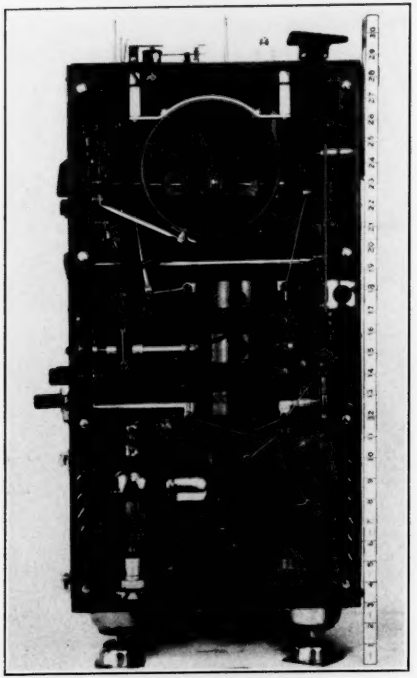


FIGURE 10.

Interior View of L. F. Unit of an Obsolescent Type of High Power Aircraft Transmitter. (High Voltage Power Supply

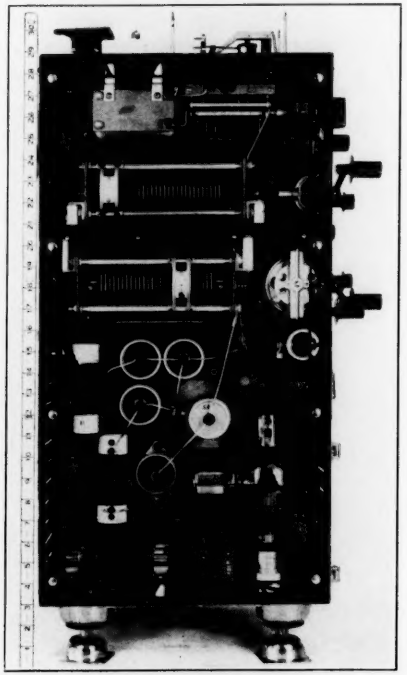


FIGURE 11.

Interior View of H. F. Unit of High Power Aircraft Transmitter of Same Type as Shown in Figure 10. (Power Unit

versely proportional to the square of the frequency: $L = \frac{25330}{C_a f^2}$, where L is in microhenries, C_a in micromicrofarads and frequency is in Megacycles.

Furthermore, the equivalent antenna capacitance, C_a does not remain constant with frequency but varies as indicated in Figures 4, 5, and 6 of the first part of the article, becoming smaller as the frequency is lowered, and hence requiring more and more inductance to resonate the antenna.

Both effects necessitate greater spacings, more and higher-grade insulation, larger and heavier capacitors and resistors, and increased clearances to grounded members in order to avoid high-voltage flash-overs and corona discharge. Note the large inductances, large resistors, capacitors, and heavy insulation, etc. in the high-power, low-frequency aircraft transmitter unit shown in Figure 10. Contrast this with the high-frequency transmitter unit (same power) of Figure 11. In this case, the size and weight (46 pounds approximately) of each unit are identical but the latter unit covers many times the frequency range of the former.

Interior View of H. F. Unit of High Power Aircraft Transmitter of Same Type as Shown in Figure 10. (Power Unit [illegible])

Interior View of L. F. Unit of an Obsolescent Type of High Power Aircraft Transmitter. (High Voltage Power Supply [illegible])

It should be recalled now that the dielectric strength of air is proportional to the air density and therefore varies directly as the pressure and inversely as the Absolute temperature. In the atmosphere the air pressure is normally one-half the sea-level value at about 18,500 feet altitude and becomes one-quarter sea-level pressure at approximately 35,000 feet, while the normal temperature gradient is a reduction of one degree Centigrade for each 500 feet of altitude on the average. At high altitudes the temperature remains fairly constant at low values. The net result is a proportionate reduction of air density with increased altitude, the combined effect of temperature and pressure being simply to displace the Density versus Altitude curve to the right of the Pressure-Altitude curve. The average relative humidity under normal conditions also decreases with altitude, becoming very low at high altitudes. At the lower altitudes, especially in the tropics, the ambient temperature inside a radio transmitter in a closed airplane may become as great as 65 degrees C., with surrounding relative humidities up to 100 per cent. Heat dissipation in naval aircraft radio equipment, however, is probably not

of limiting importance unless hot-spot temperatures in components rise to high values due to excessive electrical losses and poor ventilation, since most flight operation will be under conditions of comparatively low ambients. Nevertheless in small and very compact transmitters of medium power output an electric fan often must be provided to prevent excessive heating of components—the weight increment becoming a necessary sacrifice to reduced overall size. In this connection, the use of iron dust cored transmitter R. F. coils (to reduce coil sizes) tends to restrict ventilation and may result in serious overheating of the coils under certain conditions.

At a given air pressure, the air gap necessary between two metal surfaces to avoid arc-over increases as the frequency increases, and the effect of sharp edges and curved surfaces of small radii toward reducing the breakdown potential also increases with frequency. The breakdown voltage per inch of air gap is consequently assumed to be somewhat less in high-frequency design than for the lower alternating current frequencies, although a breakdown voltage of 15,000 volts (r. m. s.) per inch at sea level between moderately well-rounded smooth surfaces may be safely assumed. The same considerations hold reasonably true for corona loss or brush discharge, providing the intervening insulator is dry. Both corona and flash-over are dependent upon the surface conditions of the conductors involved as well as the air density, and are influenced by the presence of neighboring conductors and to some extent on humidity. The sparking potential of dry air at standard sea level pressure (760 mm. H_g) and 25 degrees C. temperature is approximately 30,000 volts (max.) per centimeter of gap for smooth metal conductors of large radius of curvature and at commercial A. C. frequencies, although the air begins to ionize when the electric field intensity is in the neighborhood of 13,000 volts per centimeter and this ionization progressively increases as the density of the electric field is increased until at around 30,000 volts per cm., depending upon the size and distance apart of conductors, corona is continuously maintained and finally the air space is completely disrupted and a spark or flash-over occurs. The condition of well-rounded surfaces is seldom obtained in aircraft radio equipment owing to size limitations so that the sea-level corona voltage may be considerably less

than the above value. Thus, small sharp irregularities of only a few hundredths of an inch in height may reduce the voltage required to start corona by as much as 50 per cent of the value for a smooth conductor. Between needle points at standard air and 80 per cent relative humidity, the sparking voltage for continuously applied sine wave voltages of commercial A. C. frequencies is approximately 20 Kilovolts per inch, and for high radio frequencies and short duration impulses may be reduced by 25 per cent or more from this value.

When an insulator is covered with a film of either fresh or salt water, as would often be the case in operations over sea water and as a result of condensation caused by varying temperature and humidity while gaining or losing altitude, etc., the degree of voltage breakdown may be divided into the three following classes:

(1) Visually the insulator appears to "steam" or "smoke"; i. e., the losses on the surface of the insulator are sufficient to vaporize the water but the voltage is not sufficient to produce either visual corona or flash-over. In a short time the insulator becomes sufficiently dry so that power is no longer dissipated in the insulator. Considerable power is lost while the insulator "smokes."

(2) Where there is corona or brushing at the high voltage end of the insulator. This voltage is not sufficient to cause actual flash-over or break-down. If the voltage is applied for a considerable length of time the insulator will become partially dry and the corona will cease.

(3) The voltage is sufficient to produce flash-over. In this case the transmitter power output drops and practically all of the R. F. power is dissipated in the arc.

In all three cases considerable power is dissipated in the insulator.

The corona or flash-over voltage is slightly lowered by high relative humidity. When an insulator is covered with a salt water film, break-downs occur at potentials between 1 and 5 peak Kilovolts and this phenomenon is practically independent of the size of the insulator. Under these conditions the radiated power may be reduced to practically zero. Tests on smooth round 1½ inch diameter pedestal insulators of high-grade porcelain with smooth,

rounded metallic caps and bases have shown flash-over to occur at a potential gradient of approximately 21 Kilovolts per inch of insulation when dry, about half this value when wet with fresh water, and one-third to one-fifth when wet with salt water. These values also apply to corrugated pedestal insulators if the length of insulation is measured linearly and not considered as the actual total outline length of insulating path, the corrugations apparently contributing little or nothing to the disruptive strength. These tests were made at standard air pressure and temperature and at a frequency of approximately 200 Kcs. Typical test results on bowl type high-grade porcelain entering insulators are given below:

Size ..... inches	12	6	2-1/2
Inside diameter of metal ring ..... do	10-3/8	5-1/2	2-1/4
Diameter of inner conductor ..... do	None	3/8	1/4
Radio-frequency measurements in peak Kilovolts:			
Insulator dry: Flash-over	> 48.5	*29	*19.3
Insulator at 100 per cent relative humidity: Flash-over	—	—	17.5
Insulator wet with fresh water:			
"Smokes"	20	—	—
Corona	> 48.5	—	8
Flash-over	> 48.5	*29	17
Insulator wet with salt water:			
"Smokes"	—	5	0.7
Corona	43	7.4	—

At the higher R. F. powers provided by aircraft transmitters employed in large patrol planes, the radio-frequency voltages may attain values as high as 15,000 volts (r. m. s.) on antennas of small capacitance, the highest voltage being obtained at the lowest frequency. Peak transmitter amplifier circuit voltages may be as high as 5000 volts. Even with relatively large size equipments, flashovers frequently occur at altitudes above 18,000 feet across lead-out insulators, coil forms, air tuning capacitors, and frequency band change switches when operating at full power. The significance of all this is that conductor sizes must be large,

* Flash-over occurred from the inner conductor to the metal ring.

wire diameters increased, amount of insulation increased, tuning capacitor and band switch spacings and overall dimensions enlarged, etc., to prevent corona and breakdown even at medium high altitudes for the maximum transmitter output powers employed. The necessity for increased size, and therefore weight, likewise applies to power cable connection plugs and receptacles, and to the antenna relay used for switching the antenna from transmitter output to receiver input and vice versa. Transmitter insulation must withstand flash-over without carbonizing, burning, flow, etc., so that in the present state of development, for both mechanical and electrical reasons, it has been the practice to employ insulation of high density such as ceramics and "mycalex" in aircraft transmitting equipments even though the weight per cubic centimeter of such materials is greater than that of many of the aluminum alloys employed in the fabrication of the equipment. Note the generous clearances and large amount of vacant space in the radio-frequency transmitter unit covering low frequencies shown in Figure 10. Also, the long operating control shafts, large frame, etc.

It may be seen from the above that the actual radiated power output is definitely limited by the antenna constants, the lowest frequency required, the maximum altitude, and size restrictions.

For a given size of transmitter, and a specified lowest frequency, there is accordingly a practical limit to the maximum power output available from the transmitter itself which is dependent upon maximum ceiling and such other operating conditions as whether a trailing wire antenna may be used, or the equipment is in a supercharged cabin or compartment, etc., and if progressive, intelligent engineering design technique is employed, this establishes the weight of the unit. It is interesting to note that where a wide band of transmitting frequencies which includes medium or low frequencies is required, an actual excess of radio power is automatically available and provided at the higher frequencies.

Fortunately, communications over medium distances at sea are more favorable at high altitude than near sea-level as a result of the direct, optical paths for propagation attainable and this fact somewhat compensates for the necessity of decreased power at the upper altitudes.

As in the case of the D. F. loop coil mentioned previously, the low-frequency antenna loading coil may be decreased in dimensions without impairing its Q and possibly with an improvement of Q by the employment of a suitable powdered iron core having high permeability. In one instance, at a given low frequency, the employment of an iron core loading coil reduced the volume necessary by 5 times below that of an air core coil of the same inductance but increased the weight by 45 per cent. The diminished size allows greater clearance to grounded parts and shielding in the space available, which in itself contributes toward increased Q, and a larger coil Q means reduced losses in the inductance, although weight is increased thereby. Until recently it has been common practice in this country to operate the higher power aircraft transmitters by local control and with short interior leads to the antenna lead-in insulator, thus minimizing radiation losses, due to shielding by the fuselage, and the stray shunt capacitance of the transmitter lead to nearby structural members and parts grounded to the fuselage which bleed off a portion of the R. F. current otherwise available to the antenna. The magnitude of R. F. current shunted through stray capacitances increases directly with frequency and at the highest frequencies of a transmitter having wide H. F. coverage may result in excessive heat losses.

In many cases, the L. F. antenna loading inductance has been constructed as a separate unit which can then be mounted close to the antenna lead-out insulator. (See Figures 12 and 13). This arrangement permits the size of the transmitter proper to be reduced. With medium power, locally controlled transmitters, it has often been the practice in Europe to place the loading coils in the tail section, where space is not at such a premium as in more crowded sections, using remote switching and tuning to obtain proper coupling and matching to the antenna (s), and to feed this unit by means of a relatively low-voltage transmission line extending from the transmitter. This arrangement may afford the maximum possible amount of power to the antenna but at the sacrifice of added weight. With the unit shown in Figures 12 and 13, the maximum low-frequency power output at high altitudes is limited by corona and flash-over owing to the small volume of the unit. Its weight is approximately 3 pounds. Had it been incorporated within the transmitter unit, the size of the

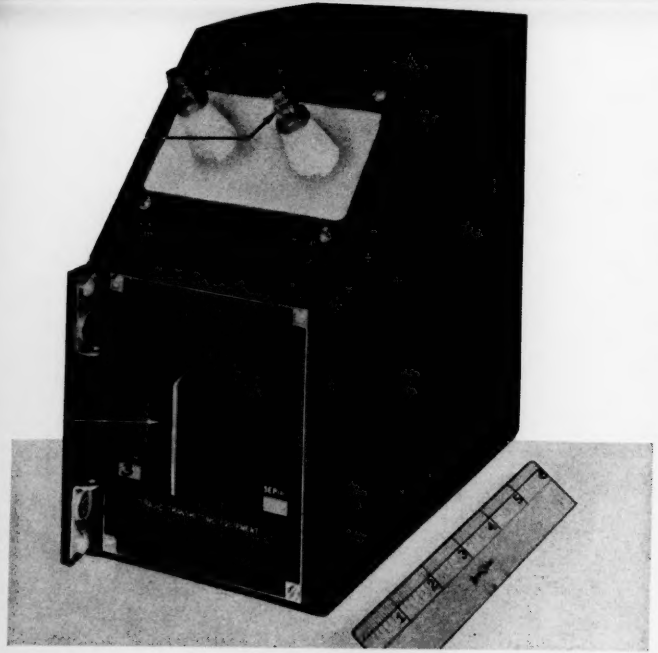


FIGURE 12.

Antenna Loading Coil—Right Front Oblique.

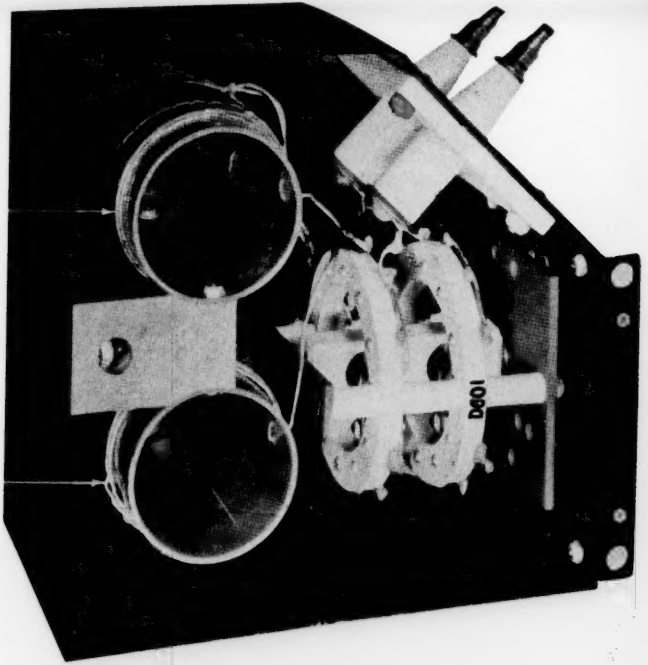


FIGURE 13.

Antenna Loading Coil—Cover Removed—Left Rear Oblique. Density = 0.014 pounds per cubic inch.

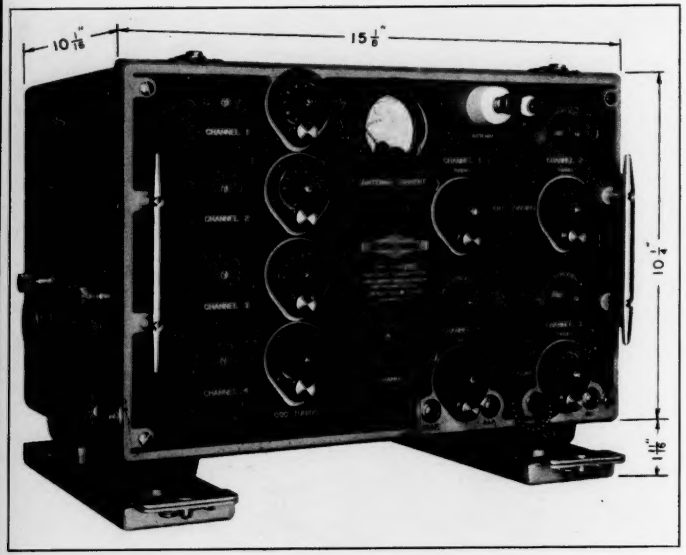


FIGURE 14.

Commercial Version of a Military Aircraft Transmitter Providing 4 Pre-set Frequencies. This covers both Low and High Frequencies and is an excellent performer for its size and weight of approximately 36 pounds. Density = 0.0195. CW, MCW, and Telephone are provided, with either Local or Remote control. Frequency range = 300-600 Kcs. (with separate Antenna Loading Unit) and 2.5 to 12 Mcs. Nominal Power Output = 60 Watts. High Voltage (dynamotor) Unit, (not shown) including filter, weighs 25 pounds. Its Density = 0.043 pounds per cubic inch (approx.).

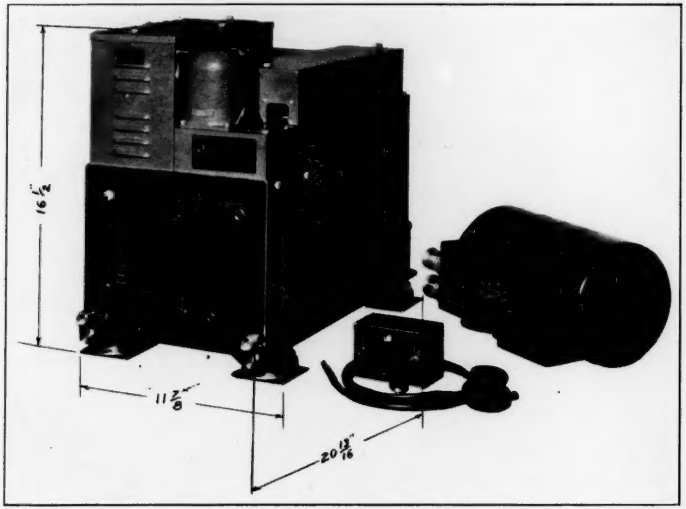


FIGURE 15.

This transmitter provides up to 10 pre-set frequencies in the range 3-10 Mcs., automatically selected from the remote control unit shown. Three additional, fixed, crystal-controlled frequencies are also provided. Adjustment to operating frequencies is automatic, electrically-operated. The equipment may be adapted for continuous tuning over the frequency range and manual operation of the autotune system. Power output = 100 Watts on both Telephone and Telegraph. Weight of Transmitter, as shown = 51 pounds. Density = 0.0125 pounds per cubic inch. Dynamotor Unit weighs 34 pounds.

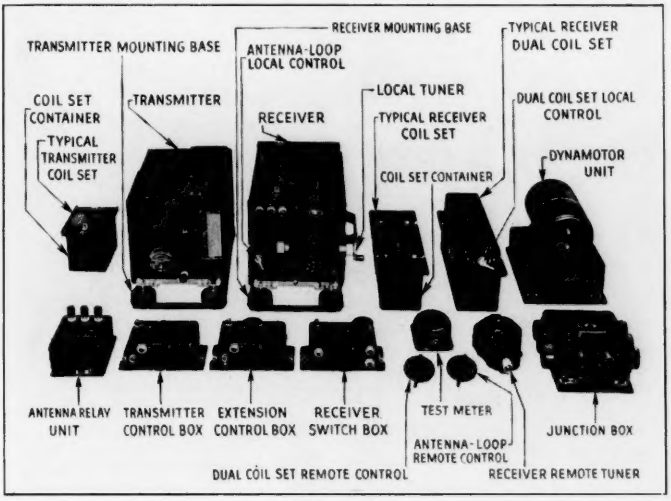


FIGURE 16.

Principal Component Units of Low-Power Aircraft Radio Equipment (less cables, etc.) of a Type Employing Plug-in Coils for Changing Frequency Bands.

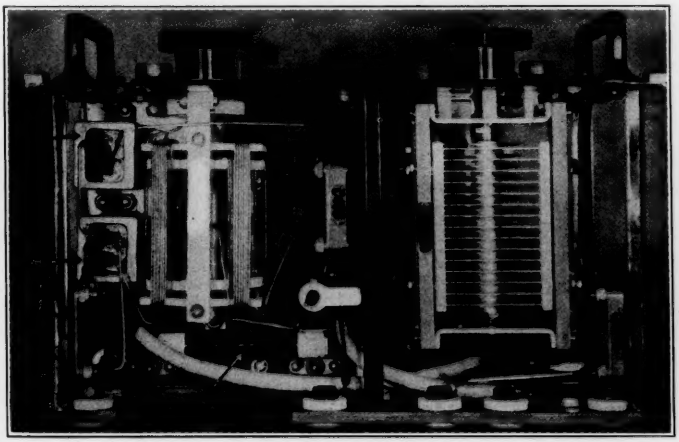


FIGURE 17.

An H. F. Tuning Unit—Covers and Shield Removed—Top View. From a medium-power aircraft transmitter. Its weight (8 lbs., approx.) indicates that considerable attention has been paid to the Weight Factor. Density = 0.015 pounds per cubic inch (approx.).

Watts. High Voltage (dynamotor) Unit, [illegible] weighs 25 pounds. Its Density = 0.043 pounds per cubic inch (approx.). 34 pounds.

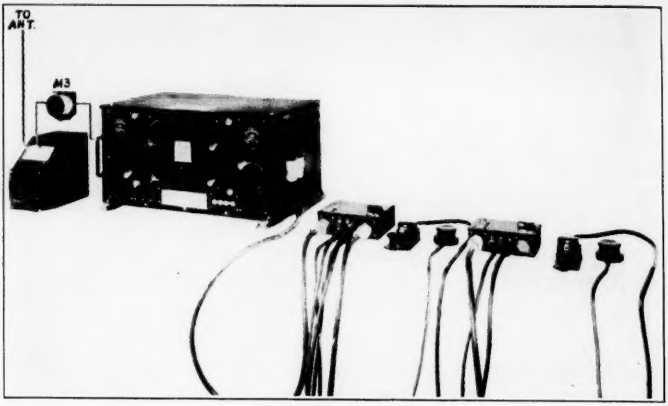


FIGURE 18.

Medium-Power Naval Aircraft Radio Transmitting Equipment— assembled for operation.

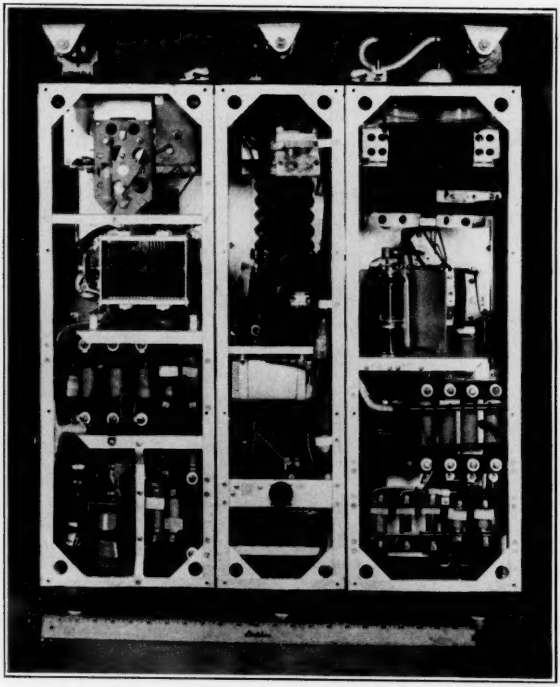


FIGURE 19.

Rear View of an assembled High Power Aircraft Transmitter covering both Low and High Frequencies. Note the heavy transformers, etc., in the Power Unit and the large, heavy, ceramic-insulated R. F. coils in the low frequency portion. Overall Density is approx. 0.009 pounds per cubic inch.

latter would have been greatly enlarged and the total weight increased considerably.

Figure 9 shows the high frequency portion of an R. F. transmitter (not including high-voltage power supply) which has a total frequency coverage practically the same as that of the H.F. unit shown in Figure 11. Power outputs are very nearly the same for both. The weight of the unit shown in Figure 9 is approximately 25 pounds as compared with 46 pounds for that of Figure 11.

In the case of the larger unit, its size has been mainly determined by considerations of symmetry and convenience of coupling the L.F. and H.F. units of the composite transmitter together, with a power unit of same height and depth in between, rather than on a size and weight basis.

Following the provision of low frequencies in the equipments, taken together with the tendency toward increasing transmitter powers due to the need for communication over longer ranges and as a result of reduced antenna efficiency, the inclusion of a wide range of operating frequencies constitutes the next most severe requirement from an equipment weight standpoint, especially because of the usual demand for continuous coverage with but little if any hiatus. On the other hand, however, where the lower end of the radiofrequency range to be included in the apparatus is not much below 3 Mcs., a reasonably wide range of frequencies with a choice of several discreet operating frequencies and provision for their rapid selection may result in less overall weight than otherwise since less power is needed to establish satisfactory communications when a choice of any frequency or of several frequencies distributed over a moderately wide band is available. The transmitters pictured in Figures 14 and 15 are good examples of equipments embodying these features. In other words, a reasonable amount of power on the right frequency is far better than a large power output on a wrong or unusable frequency. For on frequencies above a transition region somewhere in the neighborhood of 2000 Kilocycles, an enormous amount of power may be of little value and often totally ineffectual toward establishing communication if the operating frequency is incorrect for the time of day involved, the season, distance between transmitting and receiving stations, and noise level at the receiver on

that frequency. This situation often occurs, for example, when the receiving location happens to lie within the "skip distance" for the particular frequency used or in a zone of deep and frequent signal fading.

It has heretofore been the practice, at least in this country, to employ plug-in tuning units for both receivers and low and medium- power transmitters in naval aircraft. (See Figures 6, 7 and 16.) In the case of receivers, these have been largely of the tuned-radio-frequency type and consequently the plug-in units have consisted simply of coupled R.F. coils or transformers with only their associated resistors and by-pass or trimming capacitors plus the required shielding, connection plug and fastening arrangements; each coil set being capable of covering approximately a 2:1 frequency spread in conjunction with the main ganged variable tuning capacitor located in the receiver proper. A typical receiver coil set of this type will seldom weigh much more than 2 pounds. The total frequency range of such a unit may also be split up so that part of the coverage lies in one band of the radio-frequency spectrum and the remainder in a band somewhat separated from the other. In this case a switching mechanism must additionally be supplied with a consequent increase of weight, particularly when remote control is necessary. Somewhat similar arrangements, except for the provision of band switching, have applied to low-power transmitters. However, for medium power transmitters, the removable tuning units have not been as simple as this but have of necessity contained almost the entire gamut of components required for tuning and other R.F. adjustments and each includes the variable air tuning capacitor or variometer, variable inductances and their associated spacious, low-capacitance switches, and other circuit elements. (See Figure 17). The band of frequencies which can be confined within a single unit of small enough size to permit installation of the transmitter in any but large patrol planes is narrow owing to the R.F. power and voltages involved and the rapid falling off of efficiency as the range is extended, although such units may weigh as much as 16 pounds each, depending upon the actual frequencies covered. Consequently, it may be seen that if aircraft must cover a wide range of transmitting and receiving frequencies the weight goes up enormously and in direct proportion to the coverage required.

For a continuous, total frequency spread of 8 times the lowest frequency to be covered, a minimum number of from 3 to 6 plug-in units (depending upon the specific lowest frequency included) must be carried both for the transmitter and receiver. In addition, metal carrying cases or protective stowage containers must be provided for each of the units not actually in operation. A medium-power transmitter of the plug-in tuning unit type and obsolescent design is shown in Figure 18, complete with antenna loading coil unit and remote control boxes.

For high-power and some medium-power transmitters it has been customary to build all components either into a single large unit or in separate sections having the same height and depth and coupled together side by side both mechanically and electrically to form a single composite unit as installed. (See Figure 19). This latter arrangement becomes requisite for many installations in order to permit passage of the equipment through the small size hatches ordinarily provided in the fuselage. In these equipments, removable or plug-in units covering discrete frequency bands are not employed as an operating procedure so that the inductance coils and capacitors for the entire frequency range to be accommodated must be built in as more or less permanent fixtures and carried at all times. This obviously results in large dimensions, accompanied by the maximum of weight. Radio-frequency band switches for rapidly changing from one tuning range to another must be included, and these must be designed with generous amounts of supporting insulation, spacings between contacts and between switch members, and shielding or other grounded conductors in order to minimize stray capacitance, which would restrict the tuning ranges and increase R.F. losses, and to minimize high-voltage corona and flash-over.

It must be remembered, also, that for the most part these considerations apply in multiple to radio transmitters suitable for Navy use since they must be multi-stage in order to achieve acceptable frequency stability. Where reasonable frequency stabilities are to be afforded, the master oscillator-power amplifier type of design must be resorted to even when crystal-controlled oscillators are employed, and at least one isolated amplifier stage and its coupled circuits must be used in addition to the frequency-determining tubes and circuits. This is especially true in aircraft

equipment wherein the output power-amplifier stage must work efficiently into widely varying antenna constants both on aircraft of different sizes or types and between antennas on a single airplane. Due to this fact, plus the relatively small size antennas available on most naval aircraft, and because the fixed antenna constants are different when the craft is close to or on the ground, carrier deck, or water than when aloft, considerable flexibility in the form of tuning elements and circuits, interposed between the antenna terminal and power amplifier, must be provided to permit proper coupling to the antenna load and ready tuning of the antenna to resonance while in flight as well as to electrically isolate the output from the master oscillator or intermediate tuning circuits and prevent detrimental reaction and interaction. It will be noted, by reference to Figures 5 and 6 in Part 1 of the article (November, 1940), that with the larger fixed antennas on patrol planes, etc., capacitive antenna circuit tuning must be employed for frequencies at which the antenna reactance is inductive and that inductance loading again becomes necessary as the frequency is further raised. Although the additional circuit elements may not contribute seriously to the transmitter compartment size, since the required values of inductance and capacitance are smaller than for lower frequencies, they account for their share of both size and weight. It may be observed from Figure 20 how wide frequency coverage adds to the bulk by requiring a greater number of coils, switches, etc., to provide the extended range. This incidentally, represents an exceptionally compact design.

When the frequency range is appreciably extended toward higher frequencies the stray capacitance and dielectric losses increase and less power is available as useful radiation; also, additional vacuum tube stages must be employed as frequency multipliers. For a given tube and tube input power there exists an optimum frequency range as regards power output and efficiency. Indicative of this trend is the table shown in Figure 21—applicable to small, aircraft type transmitters. As the degree of frequency multiplication increases, tube efficiencies fall off rapidly and in order to maintain an effective radiated power, tube voltages must be increased and/or a greater number of tubes employed, electrical ratings of all parts must be increased, and the total

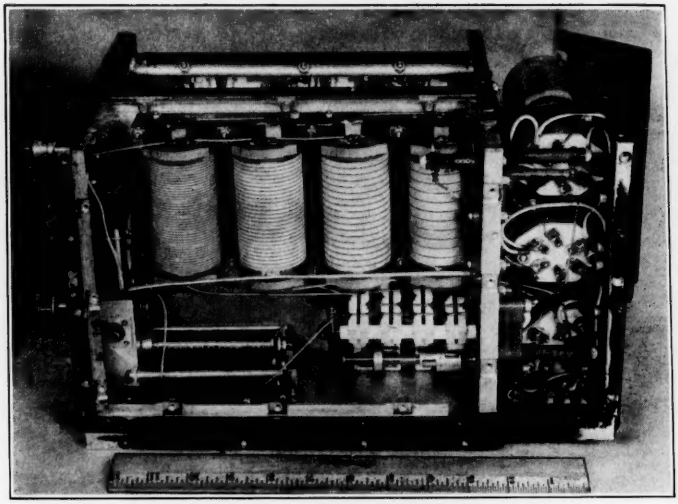


FIGURE 20.

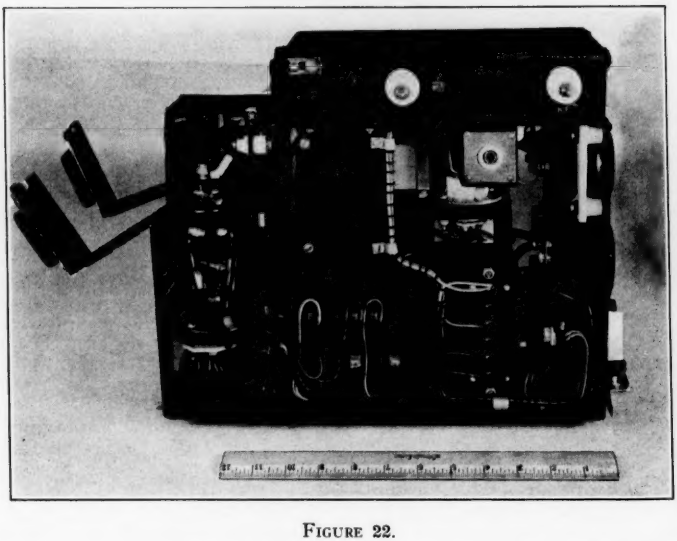


FIGURE 22.

A compact High Voltage Power Unit of a moderately high-power transmitter. Its Density is approximately 0.028 pounds per cubic inch.

result is an overall increase in power requirements, expansion of size, and much added weight. The weight influence of the H.V. power supply may be appreciated from the fact that the weight of the Power Unit shown in Figure 22 is approximately 40 pounds and accounts for practically one-half the total weight of the entire transmitter. The problems of accuracy in rapidly and conveniently setting up to a correct frequency and of frequency stability at very high frequencies require increased and sometimes involved precautions and features which add to both volume and weight. In larger transmitters which are subdivided as mentioned above,

FREQUENCY	POWER OUTPUT (CW)
2500 KCS.	15.5 WATTS
3500 KCS.	17.0 "
5000 KCS.	17.5 "
6000 KCS.	17.5 "
7000 KCS.	16.5 "
8000 KCS.	16.0 "
10000 KCS.	12.0 "
12000 KCS.	14.0 "
14000 KCS.	11.5 "
16000 KCS.	7.0 "

FIGURE 21.

the low frequencies are usually built into one section and the high frequency range in another with a high-voltage power supply in between. In this case, if the low frequencies are dispensed with and its section removed, the remaining equipment would, to a great extent, be excessively powered and over-weight because of the excess power equipment and higher ratings required for low frequencies. The result of providing a wide frequency range including both low and high frequencies in a single transmitter, even if subdivided, is that the H.F. section or portion of the equipment possesses a substantially wider frequency coverage and greater output power than the L.F. portion in the same volume and weight.

In more recent designs, one approach has consisted of construction of individual, self-contained units mounted either separately or in groups, with each unit covering a discrete frequency band generally having a tuning ratio not over 2:1, and with a separate modulator and interphone unit. This system affords an efficient, compact design of reduced size and weight per unit, but where continuous, wide frequency coverage is demanded, the total installed weight rises proportionately, and extra weight must be carried owing to the duplication of many parts and functions where necessary, together with the added amounts of interconnecting cables, control linkages and attachments The system nevertheless possesses the distinct advantages of great flexibility, providing separate, multiple channels which may be adjusted to several desired frequencies and selected at will or operated simultaneously; and furthermore, if a restricted range of frequencies suffices for a given flight, the minimum of weight and equipment may be carried, and the extra weight of the remaining complement of unoperated units left on the ground or on board the aircraft carrier where it belongs. The same reasoning largely holds true for the plug-in coil or tuning unit type of construction but with the important difference that simultaneous, multi-channel operation is not provided by the latter unless the entire equipment is duplicated, in which event the combined bulk and weight would be prohibitive for most aircraft.

This procedure also permits installation of units mounted separately in whatever restricted spaces are available for radio in the airplane—a great advantage where space is at a premium, and one which may often determine the type and quantity of equipment to be installed, regardless of any added weight increment over the minimum possible.

The single band, individual R.F. transmitter or receiver unit type of construction provides a limited frequency range per unit and therefore permits a higher electrical efficiency and maintains a more constant efficiency over each band for a given size that is obtainable in one equipment covering all bands and frequencies. One version of this type of construction is illustrated below (Figure 23).

Such a unit has its own high-voltage dynamotor power supply integrally mounted. Even so, the components in small, low power

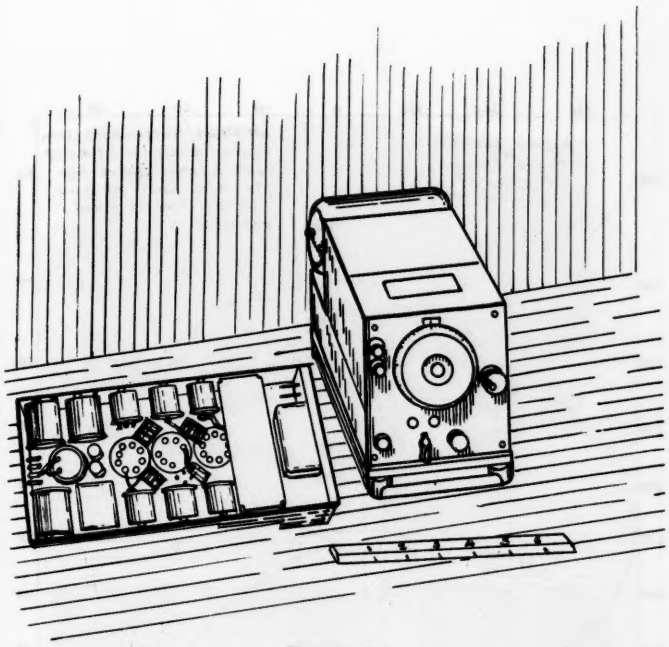


FIGURE 23.

equipments cannot be too compressed or crowded dimensionally or spurious oscillations and harmonics will occur.

Another and seldom recognized consequence of covering a wide continuous frequency range results from the large variation of antenna directivity characteristics with increasing frequency, the space pattern becoming more and more distorted and containing a larger number of nulls as frequencies are used which are sufficiently high to make the electrical length of the antenna an appreciable fraction of a wavelength and longer. The resolution of this effect into terms of added weight is indirect but nonetheless real. Thus, when the transmitting and receiving stations are so positioned that either or both lie within a null sector of their antenna patterns at the frequency employed, satisfactory communication can be established and maintained only with great difficulty, and under these conditions considerable increase of

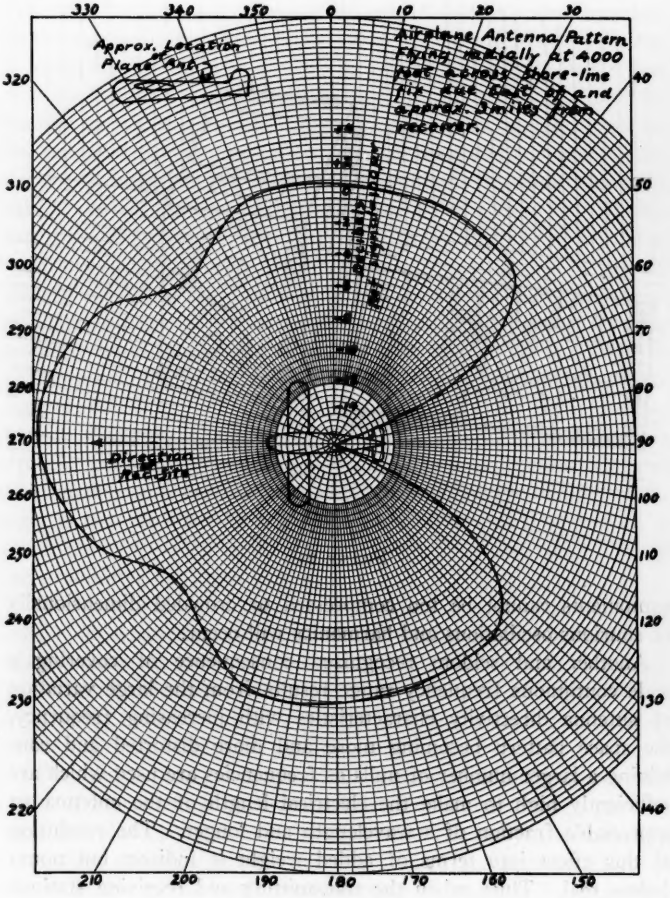

330
340
350
0
10
20
30
320
40
310
50
300
60
290
70
280
80
270
90
260
100
250
110
240
120
230
130
220
140
210
200
190
180
170
160
150
Approx. Location Plane Ant.
Airplane Antenna Pattern. Flying radially at 4000 feet across three-line fix due East of and approx. 3 miles from receiver.
Direction Receiver

FIGURE 24.

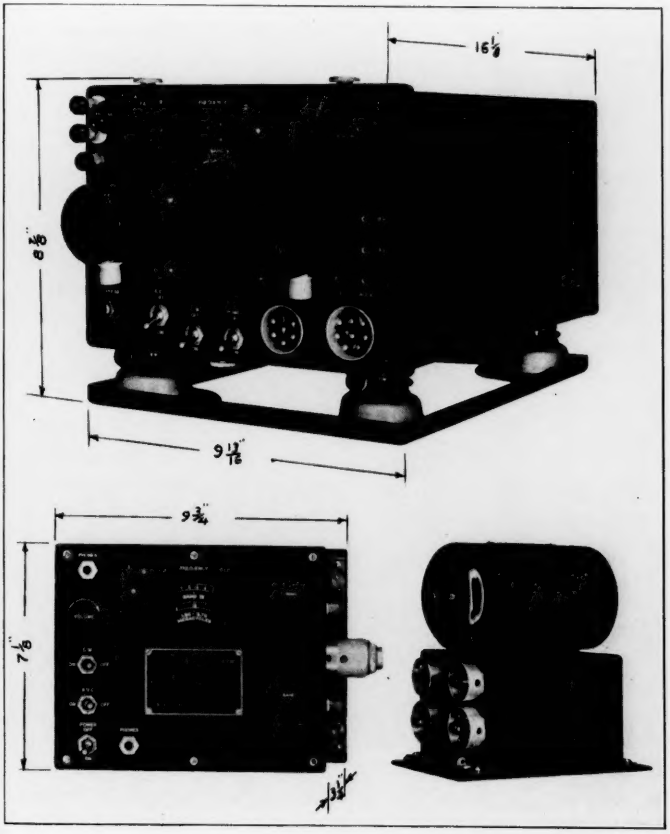


FIGURE 25.

Weight of Receiver Unit, as shown = 26.5 pounds. Density = 0.019 pounds per cubic inch.

Weight of Remote Control = 4.25 pounds. Density = 0.019 pounds per cubic inch.

Weight of Dynamotor Power Supply Unit = 8.9 pounds.

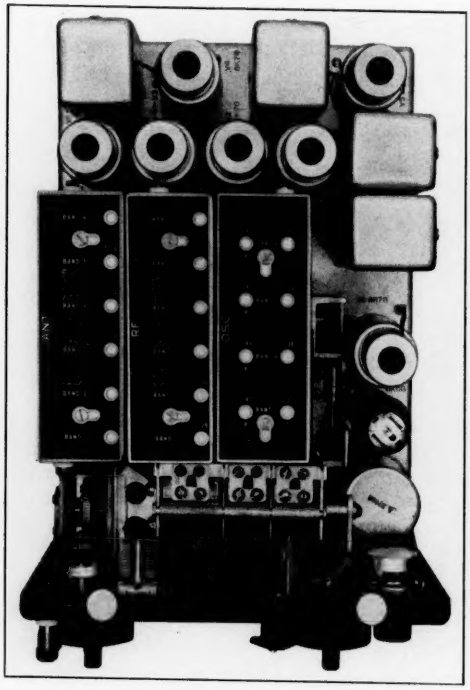


FIGURE 26.

power would be required to provide usable results. This situation may occur both between aircraft in flight and in plane-to-ground operations and is almost solely a function of frequency, associated antenna pattern, and the vertical and horizontal angles between the transmitting and receiving locations. Figure 24 shows a representative pattern of the field surrounding a vertical, high-frequency mast antenna. Accordingly, if very high frequencies (above approx. 15 Mcs.) are to be employed, a greater power rating must be built into the equipment to insure reliable communication at all times. This condition of dead spots on certain frequencies also suggests the inclusion of a number of separate operating frequencies, either pre-set before take-off or adjustable in flight, the choice of which may be rapidly selected by the pilot or operator. These features, however accomplished, add weight but any augmented weight may be justified by the increased reliability provided.

Many of the later receiver designs are of the superheterodyne type employing frequency band switching, either locally or remotely controlled, and provide continuous tuning over each of the several frequency bands required. For any usable degree of operating ease and performance each band cannot cover a frequency range greater than 2 or 3 times the lower end frequency, and therefore the number of bands included largely determines the overall size and weight of the unit. An example of this type is shown in Figures 25 and 26 which pictures a commercial version of a similar type of military aircraft receiver and designed for either local or remote control.

For a specified total frequency range wherein all frequencies must be available at all times in aircraft, this type of construction results in substantially less weight than would be the case with either the plug-in coil system or individual single-band unit type, although the unit size of the multi-band receiver will be appreciably larger. This also applies to new types of fighter equipments utilizing full remote electrical control of all functions including tuning of both transmitter and receiver and in which any one of a number of pre-adjusted frequencies may be selected by pushing buttons or manipulating switches. Although too large and heavy to be classed as fighter equipment, the transmitter shown in Figure 15 fulfills these latter requirements, as does also that shown in Figure 14.

From the receiving standpoint, high altitude operation of itself presents little difficulty because of the low voltages involved in the receiving equipment, generally not over 300 or 400 volts D. C., neglecting those R.F. or impulse voltages which may be passed along to the receiver input from induced potentials on the antenna resulting from atmospheric electrical discharges, precipitation static, etc. However, the use of higher powers in aircraft transmitters aggravate the problems of overload and automatic volume control in receivers owing to formation flying and the necessity for both intra-squadron and inter-squadron and distant base communications so that the size and weight of the receiving equipment must be increased somewhat to include a sufficient number of tuned circuits to provide adequate pre-selection and overall selectivity, and such other protection and operational precautions as necessary. For an antenna of limited size, the stray capacitance of the lead-in wire from the fuselage entering insulator and the antenna break-in or "listening through" relay to the receiver is of great consequence as is likewise the shunt capacitance of any shielding or metal conduit required for the prevention of undesired or local pick-up and excessive R.F. voltages to the receiver input, such as when the radio installation is in the transmitting condition, etc. With fixed aircraft antennas of small capacitance this becomes especially important because any stray, parasitic capacitance inside the fuselage, although in shunt with the receiver inputs, must also be considered as in series with the effective antenna capacitance and therefore the combination acts as a voltage divider with the received signal voltage delivered to the receiver input dependent upon the ratio between antenna capacitance and the sum of the antenna capacitance and stray capacitances. At low frequencies this ratio decreases due to the small equivalent antenna capacitance, and the attendant reduction in input efficiency may enormously effect the design of the receiver, particularly when reception of a wide range of frequencies is desired. Actually, the voltage which appears at the receiver input circuit is dependent upon the ratio of the antenna impedence to the sum total of the input circuit and antenna impedances and which include the antenna and input or "load" circuit resistances, the antenna and load reactances. The maximum amount of energy is absorbed by the receiver input circuit when its resistance component equals

the total resistance component of the antenna and the reactive component of the load circuit is equal in magnitude and opposite in sign to the reactive component of the equivalent antenna impedance. If a series input trimming capacitor is incorporated in the receiver to allow adjustment of the input circuit to resonance in combination with the total external shunt capacitance present, a voltage loss is nevertheless suffered in this series reactance. Thus, the efficiency of the receiver's input circuit for the transfer and utilization of induced signal voltages on the antenna is all-important to successful reception and this will vary appreciably with the total frequency range to be covered by the unit. As regards low temperatures and high humidities in aerial operations, it is a relatively simple matter to provide sufficient reserve gain (amplification) to compensate for their detrimental effects without material increase in weight. From the frequency stability viewpoint, however, this may not be true, and in addition to stability requirements, the final weight will depend largely upon the tunability, resetability, and calibration accuracy demanded. The need for low frequency reception tends to increase both the size and weight of a receiver in accordance with the lowest frequency to be employed, the range of frequencies to be covered, and the type of performance desired. The percentage increase in weight and dimensions occasioned by the inclusion of low frequencies in receivers is not nearly as large as in the case of transmitters, however, since high R.F. voltages are not involved, and small, compact coils and transformers of good Q are obtainable; the increase being brought about principally by the additional R.F. coils and transformers, the larger windings and coil forms required therefor, and the larger capacitors necessary for tuning to low frequencies, for by-passing and filtering, etc. Nevertheless, it is the expanse of frequency range to be included that usually composes the predominant factor toward weight unless a band of very low radio frequencies must be covered, since the size or bulk will increase in some proportionate relation to the frequency range to be included. For purposes of size and weight determination the foregoing discussion also applies generally to aircraft frequency measuring standards or crystal frequency indicating equipments used for checking true transmitter frequency, receiver calibrations, etc., since such equipment may be classed in the same category as receivers.

The progressive demand on the part of the aircraft designer for smaller antennas, with which low or even medium frequencies are no longer utilizable with any useful degree of efficiency, and other considerations such as overcrowding and interference in existing frequency channels and the smaller number of available channels in these frequency bands have driven the radio engineer more and more toward the use of ultra-high frequencies; i.e., above 30 Mcs., especially on the smaller, high-speed airplanes. From the aerodynamic viewpoint, extremely high frequencies may offer a welcome relief as regards drag since the antennas may be considerably smaller than otherwise, although some increased structural difficulties are involved because in many installations the vertical antenna, generally consisting of a vertical insulated stubmast, must protrude either underneath the fuselage belly or extend upward in some location where the body of the airplane will not shield the radiation or seriously distort its field pattern for both transmission and reception and therefore must be either provided with retracting or collapsing mechanism unless located at some inaccessible and remote point on the airplane. The position of the U.H.F. antenna and the overall field pattern may result in very poor communication under many conditions (See Figure 24), and no reasonable increase of transmitter power (and weight) can be expected to counteract this.

In addition to that of reduced antenna structure and some relief from the congestion at low-frequencies, many other incidental advantages accrue to the employment of ultra-high-frequencies, not the least important of which is the fact that they afford some degree of security owing to the quasi-optical behavior and limited range of wave propagation at frequencies approaching the upper extremes of the radio spectrum.

To a great extent these frequencies can replace the low frequencies without appreciable loss of useful communication range, for reasons stated previously, thus affording a weight saving in the equipment installed. A much larger percentage of the available radio-frequency power from a transmitter can be delivered into the antenna for useful radiation at ultra high frequencies than is possible at low frequencies and the overall power requirements of the equipment and its size and weight can be accordingly decreased. However, the number of channels lost by abandon-

ment of low frequencies cannot be entirely compensated for by U.H.F. operation, and this becomes more and more important as the number of aircraft, squadrons, carriers, and bases increases. With present technique the number of available channels that can be accommodated in the ultra high frequency region is limited because of the necessity for considerable gaps between channels in order to take care of the poorer frequency stabilities of transmitters and receivers small enough to be placed in aircraft and the broader selectivity of such receivers at the very high frequencies. While ultra-high-frequency operation offers a partial solution to the difficulties encountered at low frequencies it can hardly be classed as a cure-all for existing ills. Nor can it be expected that the application of ultra-high frequencies alone will decrease the bulk and weight of equipment, for although the size of U.H.F. equipment may be reduced somewhat below that for medium high frequencies owing to the smaller electrical elements generally required, equal or greater weight may be anticipated in the complete installation unless performance is greatly compromised. Here, too, continuous coverage over a wide band of frequencies is no simple problem, especially insofar as the antenna is concerned.

This is because conventional circuits and components often cannot be employed with any reasonable efficiency and stability while even with specially designed small size parts and vacuum tubes, tube efficiencies are relatively low, circuit losses are high, and operating difficulties are aggravated. To attain acceptable performance, a larger number of tubes and tuned circuits or unconventional and bulky circuit elements and tuning devices must be included in the equipments. As one example, relatively heavy, low-loss concentric transmission lines must be employed between antennas and equipments if prohibitive losses are to be avoided at the higher frequencies. In many cases it is necessary to seal the lines and fill them with a dry gas under pressure to prevent virtual short-circuit due to high humidities and moisture condensation across the internal insulation, causing the addition of various forms of gas-tight plumbing fixtures. Due to the magnified losses involved when an attempt is made to cover a wide frequency range, including antenna coupling and tuning losses, low overall efficiency at the limits of the range, and general

tuning difficulties, the incorporation of a wide frequency range with continuous and variable coverage per equipment at ultra-high-frequency is practically out of the question for aircraft.

Further, because of the working distance limitations at these frequencies, such equipment must be separate and supplementary to the medium H.F. equipment suitable for long range communication in extended naval operations and therefore can supplant the latter gear only in short-range fighters. With this exception and from the weight standpoint, extra equipment (including the antenna) means added weight. U.H.F. equipment, despite its recognized and peculiar advantages, creates new and stubborn afflictions to plague the engineer. Thus, in the escape from atmospheric static at low frequencies he encounters high engine-ignition noise levels at ultra-high frequencies and attendant shielding and bonding difficulties.

ACKNOWLEDGEMENT.

The writer expresses his appreciation to Mr. T. P. Wright, now Asst. Chief of the Aircraft Section of the O. P. M., and to the Institute of the Aeronautical Sciences for their courtesy in permitting the use of Figure 5 and certain data relating thereto which appear in this part (II) of the article.

The courtesy of the following organizations in furnishing certain of the photographs shown in this part is also greatly appreciated—Aircraft Radio Corp.; Bendix Radio Corp.; Collins Radio Co.; R.C.A. Mfg. Co.

STEERING GEAR CONTROL CHARACTERISTICS.

By A. M. Nickerson, Jr. *

The steering of vessels has received considerable study by naval architects. There is little in the literature on the subject of steering gears themselves. This article develops the basic characteristics affecting the performance of modern hydraulic steering gears employing rams operated by continuously driven "variable stroke" pumps and automatic rudder follow-up control. It also essays to review some of their effects, practical and psychological.

—(Author's introduction.)

At first glance it should appear strange that the steering gear manufacturer is usually handed identical performance specifications for large ocean going vessels regardless of the ship type under consideration. This can be rationalized only by the fact that a steerer cannot be specified on a "rate of change of course" basis. The manufacturer is therefore held to an arbitrary time limit for turning the rudder through its "hard over to hard over" arc. The requirement standardized by custom for most large vessels is, "35 degrees hard over to 35 degrees hard over on the other side, in not more than 30 seconds." No tolerances are mentioned. This specification frequently results in technicalities not tenable if inherent steering gear behavior is understood.

The "control diagram" is drawn in terms of rudder velocity with respect to rudder position in the hard over run as shown in Figure 1. Practical considerations enter to modify this diagram in the case of each individual steering gear design type. For example, rudder velocity may be increasing or decreasing during the run depending on the varying obliquity of the link or swivel connections between the rams and tiller. Exact analyses might be based on linear ram travel rather than on angular travel of

* Engineer, Deck Machinery, Bethlehem Steel Company, Shipbuilding Division, Quincy, Massachusetts.

rudder. The effects of these are secondary to the fundamentals developed herein. The intent is to present the latter in the simplest manner and for this purpose the diagram truly represents average hydraulic steering gear control response. The figure illustrates acceleration, motion at constant velocity and deceleration. Deceleration is effected automatically by the action of the rudder

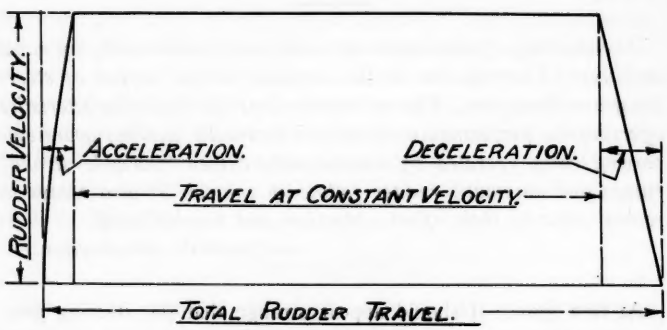


FIGURE 1.

follow-up control in restoring the pump stroke and therefore the pump delivery from its maximum value to zero.

The time required for these events is next examined. While the first period may be analyzed mathematically, an assumed time is sufficient for all practical purposes. The helmsman, forewarned of a trial run, will spin the wheel. Two seconds is a generous estimate of his ability to get things under way, especially for "ahead" steering when pressure on the rudder is an accelerating force on the whole mechanism.

The second time period is calculated from the rudder travel to the point of follow-up cutoff and the rudder velocity, which is a function of pump delivery and ram displacement.

The follow-up cutoff period offers an example of an old problem, since the instantaneous rudder velocity is proportional to the travel remaining between its position at the instant and its terminal position. In theory therefore infinite time is required to arrive at the hard over terminal. Practically, a reasonable limit may be assumed. If the rudder is considered sufficiently close to hard over

when one per cent of the total cutoff travel yet remains, this will be well within the accuracy of graduation even of large scale indicators showing ram travel as a direct measure of rudder angle. The time derivation is simplified if the cutoff travel is considered approaching zero at the hard over position and the resulting expression for time is integrated between the lower (assumed) limit and the upper at the true cutoff origin. The cutoff portion of Figure 1 is enlarged in Fig. 2, where:

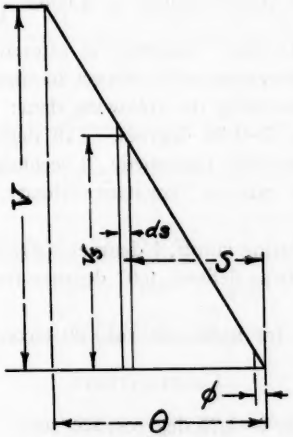


FIGURE 2.

V equals rudder velocity at beginning of cutoff. (Degrees per second).

θ is the total rudder travel during cutoff. (Degrees).

ds is an increment of rudder travel during time interval dt at position.

S, an instantaneous value of the remaining cutoff travel.

v_s is the corresponding instantaneous rudder velocity.

ϕ is the assumed limiting position of rudder for time summation.

$$dt = \frac{ds}{v_s} \; ; \; v_s = \frac{V}{\theta} S \text{ (By similar triangles).}$$

Or, $dt = \left(\frac{\theta}{V}\right)\frac{ds}{S}$

And $t = \frac{\theta}{V}\int_{\theta}^{\phi}\frac{ds}{S} = \frac{\theta}{V}\left[\log_e S\right]_{\phi}^{\theta} = \frac{\theta}{V}\log_e\left(\frac{\theta}{\phi}\right)$ Seconds.

If the above assumption is made, $\frac{\theta}{\phi} = 100$, whereby the total time for the cutoff reduces to $4.6052\ \frac{\theta}{V}$ Seconds.

A "displacement/time" diagram is offered in Figure 3 for visualization of the events with respect to time. This is based on a practical example using the following data:

Hard over run, 35-0-35 degrees. (70 degrees total).

Time for acceleration, (assumed), 2 seconds.

Normal rudder rate at "constant velocity," 3.5 degrees per second. (V).

Rudder travel during cutoff, 7 degrees. (θ).

Travel time limit as defined, 0.07 degrees from H. O. terminal. (ϕ).

Time allowance for hard over run, 30 seconds maximum.

Calculations.

Acceleration: 3.5/2=1.75 degrees/sec/sec.

Rudder travel during acceleration: 3.5/2×2=3.5 degrees.

Rudder travel at constant velocity: 70-7-3.5=59.5 degrees.

Time for travel at constant velocity: 59.5/3.5=17 seconds.

Time for deceleration: 4.6052×7/3.5=9.21 seconds.

Tabulated Results.

Event	*Seconds*	*% Total Time*	*Degrees Rudder Travel*	*Rudder Travel %*
Acceleration	2.00	7.1	3.50	5.0
Constant Velocity	17.00	60.3	59.50	85.0
Deceleration	9.21	32.6	6.93	9.9
Total	28.21	100.0	69.93	99.9

A plot of the cutoff portion of the curve shown in Figure 3 is shown to larger scale in Figure 4. Points on this curve are easily obtained for any limits desired, by the use of "semilog paper" on which space/time for this period is a straight line. (Figure 5).

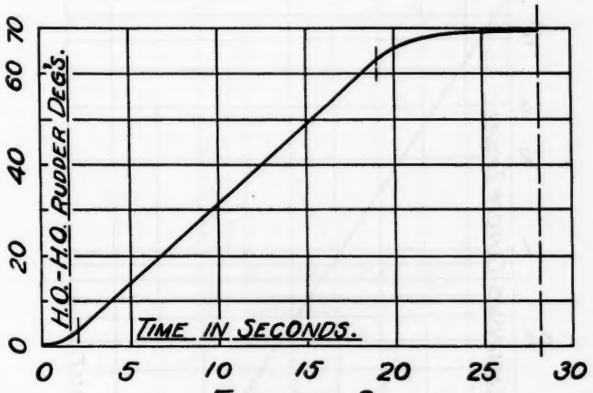


FIGURE 3.

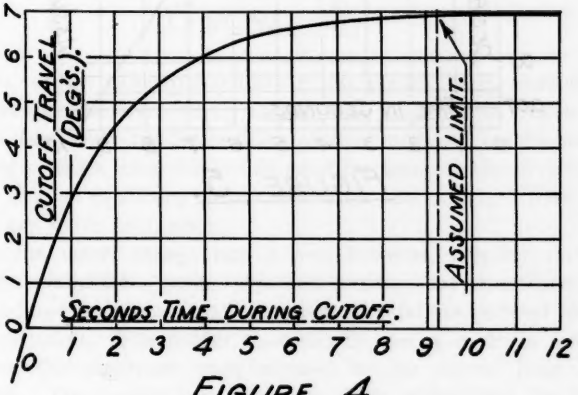


FIGURE 4.

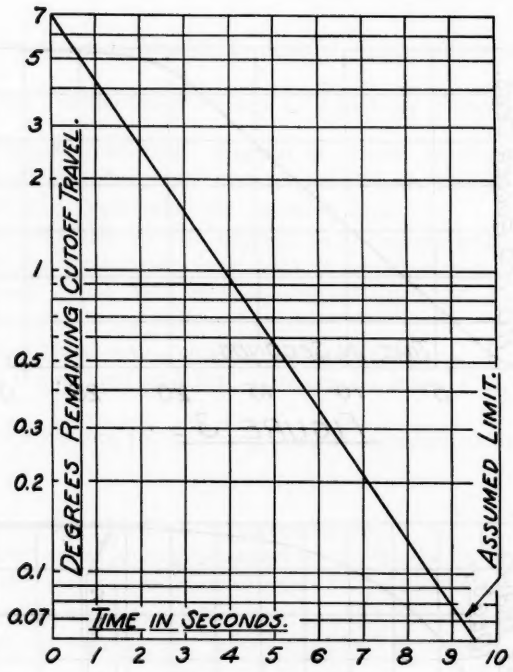

7
5
3
2
1
0.5
0.3
0.2
0.1
0.07
DEGREES REMAINING CUTOFF TRAVEL.
ASSUMED LIMIT.
TIME IN SECONDS.
0 1 2 3 4 5 6 7 8 9 10

FIGURE 5.

Figure 6 illustrates a practical factor introducing a variable in the control characteristics which is of some importance as it cannot be corrected in the practical design of the control. The figure repeats the cutoff period of the control diagram. The upper boundary represents theoretical velocity attainable at 100 per cent volumetric efficiency of the pump and no system leakage. The dash line simulates "negative velocity" due to the actual pump slip and leakage, which vary approximately in proportion to

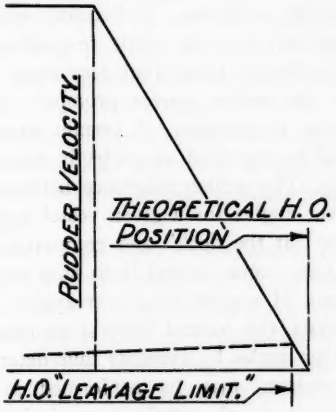


FIGURE 6.

rudder torque. The true velocity at any instant is represented by the ordinate between the upper and dash lines. This places the actual hard over rudder position at that indicated by the intersection of these lines, for at this point the pump stroke is reduced to a degree equalizing pump discharge and leakage, thus the rams can move no farther.

As the vessel swings into a turn however, pressure on the rudder diminishes, pump slip and leakage are therefore decreased and additional small quantities of fluid are pumped to the ram cylinders. The rudder consequently creeps or drifts farther toward the maximum angle allowed by the control limit stop setting. The control limit stops are those which limit the helm angles, to which the follow-up brings the rudder, and are not to

be confused with the positive stops or buffers against which the rams can take bearing in the event of a casualty. When steering astern, the effect described is reversed and the rudder moves beyond the limit stop setting. Since the leakage is increased by pump wear or service reducing the viscosity of the hydraulic fluid, (as heat in the tropics), there can be no durable prediction of the amount of drift nor any rational basis for the definition of "satisfactory drift characteristics."

The drift is normally imperceptible at small rudder angles required for holding a course. It becomes evident at the hard over angles, especially on fast ships responding quickly to the rudder, where in effect it counterbalances much of the follow-up time drag since the rudder moves promptly beyond the initial position determined by leakage. A certain amount of drift, say ¼ degree, should be regarded as perfectly normal if the steerer is in good repair. The actual difference between theoretical and practical hard over angles for a given vessel may be obtained by trying the steerer "at the dock" and comparing the angles with those obtained at the same control limit stop settings during full power runs at sea. If specific hard over angles are to be demonstrated on sea trials, the control limit stops must be set beyond the specified rudder angles by amounts thus determined.

Since ahead steering may be performed satisfactorily even though leakage prevents the attainment of designed hard over rudder angles by several degrees, we must go astern to learn the leakage limitation. At normal steering speed astern increasing leakage is remarked by a reduction in the speed with which the rudder starts from either hard over position toward amidships. Eventually the rudder will stall at hard over and cannot be returned amidships unless ship speed is reduced. If this condition develops quite suddenly it is usually due to the bypassing of fluid in the system through valves apart from the pumps, which should be examined for efficiency. However, a damaged pump can be responsible. The ram cylinder pressures ahead and astern are determined on initial trials of the vessel and the pump relief valves are set to suit the maximum, say by ten per cent over plus 50 PSI. If the pump cannot lift the relief valve when driven at full working speed and at stroke permitted by the control design, it should be overhauled.

A steering gear should be chosen as an efficient machine for maintaining the vessel's course or changing it rapidly in an emergency. Mechanically, the hydraulic steerer is one of the most efficient steering means yet devised. Of performance we may say that progress in general is reflected in the tightening of specifications or their interpretation. Thus we find emphasis laid on fractions of seconds in the time for the hard over run, and a belief that a steerer must be an instrument of precision for the attainment of exact rudder angles. These matters place an unwarranted burden on the steering gear manufacturer. An examination of the time distribution in the example quoted, and consideration of the characteristics outlined herein, should show that the end events, which are chiefly affected, have no real importance as a measure of satisfactory steering gear performance. Yet they are scrutinized by observers of steering gear trials and sometimes criticized by those who do not understand their inherence in the control type.

The observer who understands these things may "time" a trial run by watching the pump stroke control, starting his timer when the control is actuated by the helmsman, and stopping it as the follow-up returns the control to (approximately) neutral, (i. e. when there is no further motion toward neutral). In this way he is assured that the helmsman leads the follow-up throughout the run, for a pause at the wheel will add seconds to the time. He concludes by obtaining from other observers the actual hard over angles from start to finish together with the maximum power input and ram cylinder pressure, and the sum of his information is as complete and accurate as can be desired short of using recording instruments. On the other hand, the observer who is watching for definite hard over angles may be disappointed by small fractions of a degree which still are sufficient to upset his timing, as it is difficult to judge exactly when the rudder motion has stopped, due to the logarithmic cutoff characteristic.

It would seem better to disregard the end events and measure the time as the rudder passes marks determined by previous agreement. This method has some support and lends itself to a more rational specification for rudder velocity, say on the basis of a minimum rate averaged over the "constant velocity" portion of the cycle, making due allowance for the cutoff time if this rate

is derived from a theoretical time hard over to hard over. Unfortunately this method does not yet enjoy general adoption. Certainly the follow-up cutoff period is a weak thing on which to hang a specification for time. It is in fact, considered only relatively by the steering gear manufacturer, whose rates and power requirements are based on other portions of the ahead or astern travel depending on the rudder torques and detailed performance expected of the steerer.

A complete article on this subject might review the question of optimum rudder acceleration and deceleration as affected by the length of travel during the cutoff period, the relative precision or the merits thereof in different steering "telemotor" systems, and the nature of practical steering gear control systems. The scope of this paper is restricted to the simpler facts affecting hydraulic steering gear behavior. An understanding of these by all hands interested would be a distinct help to the steering gear designer.

LIGHT SOURCES FOR COAST GUARD USE.

By Lieut. Robert T. Alexander, U. S. C. G.

Although not published as Part II to the article by the author in the February, 1941 issue of the Journal, *this article is a companion to the earlier one.*

An ideal light source for a vessel of 2000 tons or less should be rugged, efficient, produce diffused light, and when installed have low brightness. It should be of such dimensions that it can be protected from direct mechanical blows and be fitted into steam and water tight fixtures. It must be reliable, easy to replace and not require special care. No single light source exists that will meet all of these requirements. However, an analytical consideration of the suitability of existing light sources will more clearly reveal the advantages and disadvantages of each type.

Tungsten Incandescent Lamps.

All installations at the present time use some form of the tungsten lamp. Three general types of tungsten lamps are available; the general service or ordinary lamp, the vibration service lamp, and the rough service lamp.

Ships (on which data is available) at present have general service lamps installed. Both rough service and vibration service lamps have been installed but, on the Coast Guard vessels considered, the failures of the general service lamp were not considered great enough to offset the added cost and decreased efficiency of the more rugged types.

Failure of the general service lamp due to continuous rather high frequency vibrations is due to the short circuiting of the closely spaced filament coils. The short circuiting decreases the resistance of the filament resulting in excessive filament temperatures and greatly decreased life. It might be said that the filament becomes excessively over voltaged to the point where the

increased lumens output is completely overbalanced by the reduced life. The vibration service lamp overcomes this by allowing the filament to sag and thus increase the spacing between coils and decrease the chances of short circuiting. Due to the more open spacing of the filament the lamp is less efficient than the general service lamp and is more liable to fail under shock.

Rough service lamps are designed to withstand shock and bumps. Failure of the general service lamp under shock is due to the fact that the tungsten filament after it has once been heated to incandescence looses its ductility and becomes brittle. The shearing stress of the shock is greater than the brittle unsupported parts of the filament can stand and the filament breaks usually close to the supports. The rough service lamp by supporting the filament at a large number of points reduces the shearing stress at any point in the filament. Shock tests conducted by the author show that with rough service lamps breakage of the bulb usually takes place before failure of the filament. However, these lamps cost twice as much as general service lamps and give 32 per cent less light per watt.

When the rated voltage of the lamp is reduced the filament becomes more rugged due to its increased diameter and at the same time the lumen per watt efficiency of the lamp is increased. Thus where low voltage (30-70 volts) lighting circuits are practicable increased ruggedness and efficiency results.

In discussing the economics of marine lighting (author's article in February, 1941 issue) the advantages of overvoltaged tungsten lamps were considered. The question may well arise as to whether the overvoltaged lamp is less rugged than a lamp operated at or below its rated voltage. It is believed that the resistance to vibration is increased by use of moderate overvoltages as outlined in the previous article. This belief is based on the fact that at higher temperatures if any change in the sag of the filament takes place it will be an increased sag with resultant increase in coil spacing thus decreasing the probability of short circuiting of coils. It is believed that these same moderate overvoltages do not produce any appreciable change in the shock resistance of a given lamp. This is believed due to greatest change in crystalline structure of the filament taking place in the early stages of incandescence.

Experimental evidence from shock tests conducted by the author

while not comprehensive enough to form the basis of a positive statement supports the conclusion that the shock resistance is not appreciably affected by moderate overvoltages. If overvoltage does reduce shock resistance then reductions in the socket voltage should produce increased shock resistance. Five general service, 115 volt, G. E. Mazda lamps, (of 100, 50 and 25 watts) operated at 117 volts were subjected to shocks of equal severity until failure of filament took place. The average lamp withstood 8.2 shocks before failure. Then six 120 volt lamps operated at 115 volts were subjected to the same shock. The average lamp withstood 6.7 shocks before failure. While with such a small number of lamps individual variations may obscure general relations, in the above test only one lamp in each group showed greater than average shock resistance and these two lamps each had almost the same shock resistance. Since the *under* voltaged lamps actually showed less shock resistance, the conclusion that resistance to shock is not *decreased* by *over* voltage seems sound.

Even if the general service lamp is rugged enough for the average service installation and efficiency is not sacrificed for ruggedness the maximum lumens per watt obtainable from tungsten lamps with optimum overvoltage is low. This is because low overhead and the present design of fixtures prevent the general use of lamps larger than 50 watts.

To obtain low brightness of fixtures and well diffused light with tungsten lamps, generally involves reductions in fixture efficiency from 15 to 30 per cent, due to absorption of light by the diffusing material. Where water tight fixtures are fitted with Navy type conical metal reflector shields decreased maximum brightness in the field of view can be obtained even with present clear glass enclosing globes by simply installing silvered bowl tungsten lamps which prevent direct view of bright image of filament and yet reduce total light output of fixture by only about 4 per cent.

The reduction of fixture efficiency to improve quality of lighting does not mean that overall efficiency or the utilization coefficient of the installation will be lowered, for with effective control of light the usable lumens on working planes, per watt, will be increased.

The small size of the tungsten lamp makes it easy to protect from mechanical injury. Perhaps the greatest advantage of the

tungsten lamp is the ease with which it can be fitted for steam and water tight installations.

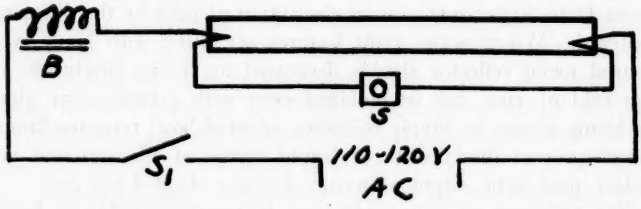
Fluorescent Lamps.

The proposal of using fluorescent lamps for marine service is worthy of consideration. Too often the novelty of a new device obscures the real advantages inherent in its design. To date installations of fluorescent lamps have been largely in the novelty class. At present installations are beginning to be made which make effective use of the potentialities of this new light source.

Ruggedness.

Are these lamps rugged enough for Coast Guard use where they must withstand not only the vibration and shock of normal marine activity, but the shock of gunfire as well? With funds provided by the Electrical Engineering Department of the University of Michigan the author undertook to investigate the shock resistance of fluorescent lamps and auxiliaries.

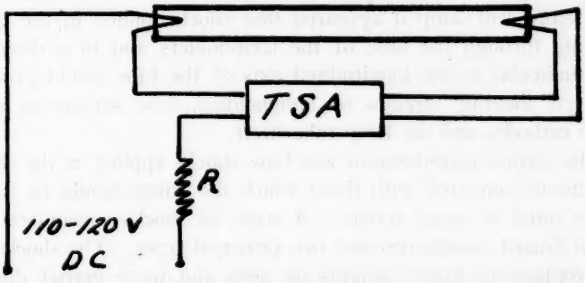
Each lamp requires two sockets or lampholders, one starter and starter socket, and an inductive ballast for A. C. operation. In A. C. installations one ballast may serve two lamps and in addition raise power factor to .95-1.00 and reduce stroboscopic effects. With one ballast to each lamp the power factor is about .60. The circuit diagram of a single tube AC installation is given below.



Each lamp has a filament at each end of the tube. When current is applied by closing switch S_1 the inductive ballast B, the filaments and the starter are in series with the line. The starter consists of two electrodes sealed in a tube filled with gas at such pressure that the line potential is sufficient to produce ionization of the gas and thus complete the series circuit. As current flows

the filaments become heated and emit electrons. The electrons are accelerated by the potential difference across the lamp and ionize the gas in the lamp. As soon as the arc discharge is established in the lamp the potential across the lamp and hence because of the parallel circuit, the potential across the starter falls to 60 volts which is below the ionization potential of the gas in the starter bulb. The starter ceases to conduct a current thus shutting off the current flow through the filaments. Positive ion bombardment of the filaments which now act as cathodes produces the continued emission of electrons by the cathodes. The ballast prevents excessive currents from flowing in the arc discharge.

In D. C. installations two lamp holders, one direct current thermal switch auxiliary and a series of resistor are required for each lamp. Only the 15 and 20 watt lamps are suitable for D. C. operation. D. C. circuit diagram is shown below.



Consider the effect of vibration upon each part of the assembly. Since the ballast is made up of a fixed inductance, and in the tulamp ballasts of inductance and fixed capacitance, placed in a waxed filled metal case vibration can have no effect on the ballast. Examination of the starter shows that its electrodes will not be affected by vibration. The thermal switch auxiliary for D. C. contains a rugged inductance and a thermal switch sealed in a small glass bulb. The thermal switch while differing in electrical characteristics has mechanical characteristics which are not unlike the glow starter. Though failure of thermal switch due to vibration is believed improbable, vibration tests of this auxiliary assembly are believed desirable. The lamp filaments or cathodes are much more rugged than tungsten lamps and since they are

called upon to carry current only for very short periods cathode failure due to vibration is for all practical purposes a very remote possibility.

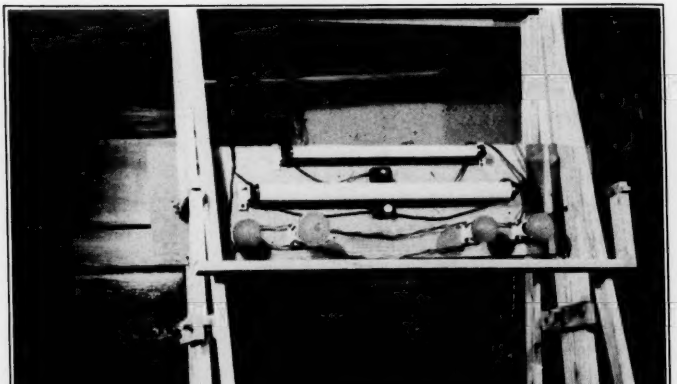
From this consideration of the physical characteristics of all parts of the lamps and their function in lamp operation it is believed that the vibration resistance of these lamps is greater than tungsten lamps. As it appeared that a vibration test was decidedly secondary to a shock test no test of vibration resistance was conducted.

Two factors were considered in setting up the shock test. First it was desired to apply the shock simultaneously to both tungsten and fluorescent lamps when both were so mounted with respect to the direction of the shock as to present the least resistance to shock. For the tungsten lamp it appeared that shock applied to the lamp socket in a plane perpendicular to the lamp axis would produce the greatest shearing stresses on both bulb and filament. For the fluorescent lamp it appeared that shock applied in the plane passing through the base of the lampholders and in a direction perpendicular to the longitudinal axis of the tube would produce greatest shearing stresses on lampholders, tube supporting pins, lamp cathodes and the lamp tube itself.

The second consideration was how shocks applied in the above directions compared with those which the lamps would be called on to stand in actual service. A study of shocks encountered on Coast Guard vessels revealed two principal types. The shock due to dropping of heavy weights on deck and wave impact due to pitching is principally perpendicular to the deck and hence to the usual mounting plane of lighting fixtures. The tungsten lamp has its greatest resistance to shock in this direction. Stresses on the fluorescent lamp would be in direction of least shear resistance for tube, supporting pins, and filament, but would give only tension-compression stresses on lampholders.

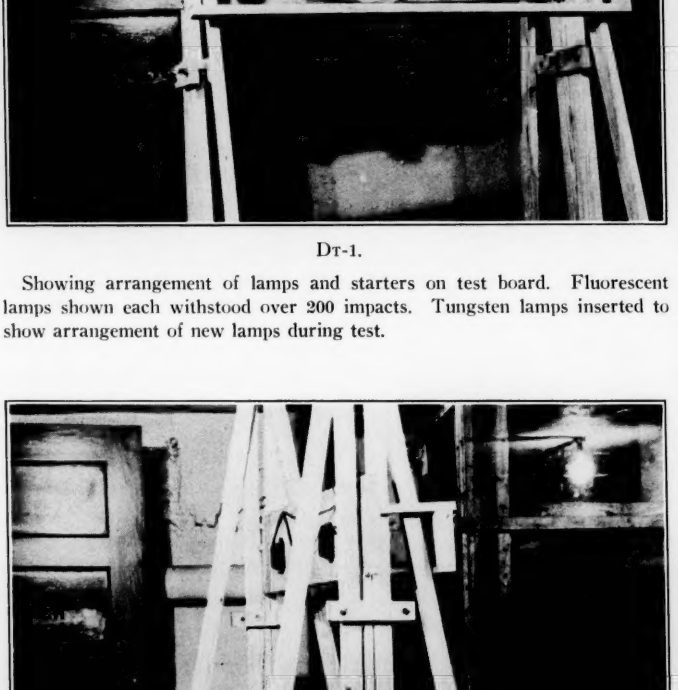
The shock of gun fire, of collision, and other horizontal impacts is parallel to the deck and hence acts in the mounting plane of most fixtures. Whether the shock acted in a fore and aft direction or athwartships, the effect on the tungsten lamp would be the same. For the fluorescent lamp the effect would depend on the orientation of the lamp with respect to the keel. If the shock were parallel to the longitudinal axis a shearing stress would

Shock Test.



Dt-1.

Showing arrangement of lamps and starters on test board. Fluorescent lamps shown each withstood over 200 impacts. Tungsten lamps inserted to show arrangement of new lamps during test.



Dt-2.

Showing ballasts on rear of test board, guides, and blocking strip.

be exerted on the lamp socket but only compressive stresses would be exerted on the tube, no stress on the supporting pins, and stress on the filament in the direction of its greatest resistance to shear.

Thus it appeared that the most damaging shocks incident to service would be fairly simulated if both lamps were arranged so that shock would be applied in the plane of both sockets and lampholders and in a direction perpendicular to the longitudinal axis of the fluorescent lamps. If shock resistance of fluorescent lamps equalled that of rough service lamps it was planned to repeat the test with the direction of shock parallel to the longitudinal axis of the fluorescent lamp. The failure of fluorescent lampholders now available under the first test coupled with lack of time caused the omission of the second test. However, when stronger lampholders are available the latter test should be made.

The test rig was in some respects similar to a guillotine. (See photographs DT-1, DT-2). Standard porcelain sockets, standard lampholders, the starter sockets and the ballasts were mounted on a yellow pine board 10 inches wide, 39 inches long and ¾ inches thick. Vertical guides were set up and a releasing gear provided so that board with all lamps in place could be allowed to drop from rest any desired height up to six feet. The test board was placed in the guides so that direction of fall was perpendicular to length and thickness of board and parallel to its width. The fluorescent lamps were mounted parallel to the length of the board. The drop was arrested by one inch square strips placed in the guides. The lower ends of these blocking strips rested on a concrete floor. Spring of the blocking strips was minimized by clamps which prevented sidewise motion of the blocking strips. The distance between points of impact of the test board on the blocking strips was 31 inches. The weight of the test board with all lamps and auxiliaries in place was 10.5 pounds. Releasing gear was adjusted so that length of board was parallel to floor throughout drop. Impact of both ends of the board with the blocking strips appeared to be simultaneous.

Average rebound of the test board was one quarter the height of fall producing a large shock closely followed by a weaker shock as the board come to rest from the rebound.

Since starter of fluorescent lamp could fail without extinguishing lamp, current was turned off and on every ten drops throughout entire shock test.

The first drop distance was 3⅛ inches which gave the board a terminal velocity of 4.11 feet per second. Four new tungsten lamps, a 100, 50 and 25 watt general service and a 50 watt rough service were in place. One 24 inch T-12 fluorescent lamp was on the board. All lamps were lighted. General service lamps were 115 volts lamps; rough service lamp was a 120 volt lamp. Socket voltage was 117 volts. Lamps were replaced with new lamps after failure. After 25 drops 2, 25 watt, 2, 50 watt and 1, 100 watt general service lamps had failed due to shearing of filament. Only rough service lamp and fluorescent lamp remained lighted. Examination of lamps, lampholder, auxiliary, and starter of fluorescent lamp showed no evidence of damage by shocks.

The test was repeated with the same drop distance and same 24 inch fluorescent lamp and rough service lamp. An 18 inch T-8 fluorescent lamp was added to board. New general service lamps of same wattages as before but rated at 120 volts were used. Socket voltage was 115 volts. At the end of 30 drops, 2-100, 2-50 and 2-25 watt general service lamps had failed. (Only one lamp survived more than four successive drops). Careful examination revealed no evidence of damage to rough service lamp or to either of the fluorescent lamps or their auxiliaries.

Drop distance was increased to 4 11/16 inches giving terminal velocity of 5.0 feet per second. New 120 volt general service of same wattages as before were installed. Same fluorescent and rough service lamps as above. After fifth drop filament of each of the three general service lamps had failed. As no damage to fluorescent or rough service lamps had been observed it was clearly apparent that the fluorescent excelled general service lamps in shock resistance, further tests of general service lamps were omitted. Another 50 watt 120 volt rough service lamp was placed on the drop board and test continued at same drop distance. On the 21st drop chipping of one of the lampholders supporting the 24 inch fluorescent was observed. The same lampholder continued to chip as test was continued but lamp remained in position and continued to operate until test at this drop distance was discontinued after 50 drops. No damage to either of rough service lamps was noted. 18 inch fluorescent lamp installation was undamaged.

Drop distance was increased to 7⅛ inches giving a terminal velocity of 6.1 feet per second. Same two fluorescent lamps and

two rough service lamps remained on board. No attempt was made to repair damaged lampholder on 24 inch fluorescent lamp. After four drops the damaged lampholder allowed 24 inch lamp to jump from holder on impact. Examination of 24 inch lamp showed that one of the four supporting pins had been loosened due to movement of lamp permitted by damaged lampholder. Damaged lampholder was replaced by new lampholder and lamp with loosened pin set in place. Lamp continued to function. The newest rough service lamp was accidentally broken. Two new rough service 50 watt lamps were installed making a total of three rough service lamps on test board. Test resumed at same drop distance. After 20 drops three lampholders for the 24 inch lamp had been replaced due to failure of the material to stand stress of shock. Supporting pin of lamp was much looser but lamp continued to operate. No damage to 18 inch fluorescent lamp installation or to any of the rough service lamps was observed. Two new lampholders for the 24 inch lamp of the "straight push" type were installed. Two drops revealed that while the "straight push" lampholders were undamaged at each impact they completely released the lamp and only luck in catching the lamp saved it.

At this point it was clear that while lampholders and lamps of the 18 inch and 24 inch fluorescent type had greater shock resistance than general service lamps in standard cleat sockets stronger lampholders would have to be provided if shock resistance equal to rough service lamps was to be obtained. Since no failure of either the fluorescent lamps themselves or of the rough service lamps had been observed it seemed desirable to attempt to reinforce standard General Electric lampholders so that a comparison of the relative shock resistance of the lamps could be obtained.

Metal and formica reinforcements were fitted to all of the fluorescent lampholders. Care was taken in the design of these reinforcements that they should not in any manner cushion the impact transmitted to the lamps. In fact it is believed that the strengthened lampholders were more rigid and actually gave a sharper shock to the lamp and its supporting pins than the original lampholders transmitted under same impacts.

Test was resumed at the 7⅛ inch drop distance. 29 drops were observed before failure of one of the strengthened lampholders for 24 inch lamp took place. Lamp was not extinguished or

released immediately upon failure of bakelite shell, but experience showed that if drops were continued brass contact strips would take a permanent set and release tube after four or five additional impacts. Damaged lampholder was replaced but after two more drops the other 24 inch lampholder failed. No damage to 18 inch lamp or lamp holders was noted and rough service lamps continued to function. Test of 24 inch lamp was discontinued although lamp and auxiliaries were functioning satisfactorily.

Drop distance was increased to 9⅛ inches giving terminal velocity of 7.0 feet per second. Test continued with 18 inch T-8 fluorescent lamp and three 50 watt rough service lamps. On the 4th drop the bulb of one rough service lamp was sheared off at its base by the shock. Fluorescent lamp and auxiliaries undamaged. After 30 drops at this distance with no further failures of either fluorescent or rough service lamps drop distance was increased.

Test continued with 11 15/16 inch drop giving terminal velocity of 8.0 feet per second. On the 21st drop the glass bulb of the first rough service lamp installed was sheared off at its base. No damage to 18 inch fluorescent lamp or auxiliaries. After the 40th drop at this distance the 18 inch fluorescent lamp although operating perfectly, was replaced with a new lamp in order to reduce possibility that an exceptionally strong lamp was being tested. After 10 more drops at this distance with no damage to fluorescent lamp and only one rough service lamp still functioning the drop distance was increased.

Test continued with 15 inch drop distance giving terminal velocity of 8.9 feet per second. On the 7th drop a slight chipping of one of the lampholders was observed. No replacements were made and test continued. On 27th drop filament failure of the rough service lamp occurred. Fluorescent lamp still in place and operating despite damaged lampholder. After the 52nd drop from this distance the test was discontinued due to failure of the drop board. At this point each impact was splintering the drop board reducing the actual shock imparted to the lamp. 18 inch fluorescent lamp and auxiliaries undamaged.

It should be noted that no fluorescent lamp was damaged so that it failed to operate in the entire test. Starters and ballasts were subjected to the same shocks and were likewise undamaged. Even with crudely reinforced lampholders the 18 inch T-8 fluorescent

lamp designed for *general service* and without special strengthening has greater shock resistance than the *specially designed* 50 watt rough service lamp. These tests lead the author to believe that with stronger lampholders the 24 inch fluorescent lamp will be found to have shock resistance equal to the 50 watt rough service lamp. The comparison with the 50 watt rough service lamp is especially significant for this lamp probably has the greatest shock resistance of any 110-120 volt tungsten lamp now on the market. An A. C. fluorescent installation was used in these tests. The mechanical characteristics of the D. C. installation so closely resemble those of the A. C. installation that little difference in shock resistance is to be expected.

Thus the facts show that these lamps with present lampholders will withstand all the vibration and shock of normal marine activity where general service lamps are being successfully used at present. With strengthened lampholders they are rugged enough to withstand any shock and vibration condition now found on Coast Guard vessels.

Efficiency.

Not only the lumens per watt efficiency of the bare source but the lumens of well diffused light emitted by fixture of low surface brightness, per watt, should be considered.

Table II.

Brightness Measurements.

Measured with Macbeth Illuminometer

Unless otherwise stated all lamps were 120 volt lamps.

		Maximum Brightness Candles/sq. in
Lovell No. 0 Ceiling Fixture,	etched spherical enclosing globe, 4¼" diameter 25 watt mazda lamp at 118 volts.	3.6
Lovell No. 0 Ceiling Fixture,	Same fixture, 50 watt mazda lamp at 118 volts. (Brightness exceeded upper range of instrument)	14.6*
Graybar Ceiling Fixture,	opal glass enclosing globe, approximated closed cylinder 8½" diameter 6" high (installed Radio room—ships offices)	
	With 100 watt lamp at 117 volts	1.55
	With 150 watt lamp at 117 volts	2.94

		Maximum Brightness Candles/sq. in
Ceiling Fixture, Commercial,	opal glass enclosing globe approximating flattened sphere. Approx. Diameter 10"—8" deep. (Installed—Engineers office)	
	150 watt lamp at 118 volts	3.12
Gray Bar Ceiling Fixture,	opal glass enclosing globe A" diameter 8½" deep approximating flattened sphere. (Installed—Wardroom—Cabin)	
	With 100 watt mazda lamp at 117 volts	.955
	With 200 watt blue daylight mazda lamp at 117 volts.	.926
Bulkhead Bracket Fixture,	Plastic diffusing shell (urea formaldehyde) approximately a paraboloid with lamp at focal point, with parchment shade of usual truncated cone shape. An I.E.S. designed. (Installed Wardroom) With 100 watt mazda lamp at 117 volts diffusing shell with shade removed	2.52
	With shade in place brightness in field of view was less than	0.2
Water tight fixtures, clear glass globes 115 volt mazda inside frosted lamps at rated voltage*		
	25 watt	18.9
	40 watt	34.3
	50 watt	41.7
	60 watt	52.3
	100 watt	98.0
	200 watt	141.0
Bare Fluorescent lamps at rated voltage†		
	15 watt, 18 inch, 1" diameter T-8	4.6
	15 watt, 18 inch, 1½" diameter T-12	3.1
	20 watt, 24 inch, 1½" diameter T-12	3.3

*Bare lamp measurements, courtesy Nela Park Laboratories, General Electric Company.

†Computed from data furnished by General Electric Company.

TABLE I

COST OF LIGHT PRODUCTION.
(Per Million Lumen Hours.)

For sources adaptable for Coast Guard use. Costs for operation neglecting initial cost of fixtures, installation, etc., computed on basis of zero labor cost of lamp replacement, for (110-120-volt) lamps operated at rated voltage, power at 2.7 cents per Kwh., lamp costs under 1939 contract.

Lamp	*Rated Life (Hours)*	*Efficiency (Lumens Per Watt)*	*Lamp Cost (Dollars)*	*Light Cost Per Million Lh.*
General Service				
Mazda I. F.				(Dollars)
25-Watt	1000	10.6	.09	2.89
50-Watt	1000	13.2	.09	2.20
100-Watt	750	16.3	.09	1.73
150-Watt	750	17.6	.21	1.64
200-Watt	750	18.2	.36	1.62
Vibration Service				
50-Watt	1000	11.0	.15	2.72
Rough Service				
50-Watt	1000	8.9	.22	3.53
Daylight G. S.				
100-Watt	750	9.8	.21	3.04
150-Watt	1000	10.5	.42	2.83
Fluorescent, white on A.C.				
15-Watt	2000	26.2	.8625	1.85
20-Watt	2000	30.4	1.15	1.65
Fluorescent, white on D.C.				
15-Watt	1000 to 2000	14.6	.8625	3.50 to 2.67*
20-Watt	1000 to 2000	20.5	1.15	2.82 to 2.10*

* Accurate data on D.C. life of fluorescent lamps not available.

Table I shows the present efficiencies of bare lamps. The wattage of the average lamp installed on Coast Guard vessels lies between 25 and 50 watts. Thus the average rated efficiency is not over 12 lumens per watt. Fundamental theory shows that present incandescent lamps have nearly reached their maximum light output per watt, so that little improvement in efficiencies of tungsten lamps is to be expected in the future. Present low wattage fluorescent lamps offer more than double present tungsten efficiencies on A. C. and about 150 per cent of tungsten efficiency on D. C. Theory of light production shows that efficiency of luminescent gas discharge tubes may be doubled or tripled by future developments.

But the bare lamp efficiency is only part of true efficiency. Because of the high brightness of tungsten lamps (Table II, 18.9 candles per square inch for 25 watt and 41.7 candles per square inch for 50 watt) to properly diffuse the light and avoid excessive brightnesses in the field of view involves considerable absorption of light by the diffusing medium. The low brightness and well diffused light emitted from the fluorescent lamps requires only little additional diffusion, hence low percentage absorption of light by the diffusing medium. In addition, the low temperatures of the fluorescent lamp permit the use of methyl methacrylate (Lucite, Plexiglas) as a diffusing medium which in providing equal diffusion absorbs less light than the thickness of opal glass or urea formaldehyde (Plaskon) necessary with tungsten lamps. Thus added to the original higher efficiency of the source is a much lower absorption factor with result that the lumens of well diffused light emitted from a fixture of low maximum brightness per watt is very much higher than with tungsten lamps.

With fluorescent lamps it is possible to produce with the economy of direct lighting, a quality of light which is equalled only by low efficiency, totally indirect tungsten lighting.

The color of light produced by the white (not daylight) fluorescent lamp has practically the same whiteness of light produced by the inefficient 150 watt blue daylight mazda lamp. (Color temperature for white fluorescent lamp 3500 degree Kelvin, for 150 watt daylight mazda, 3600 degree Kelvin).

It may appear that the fluorescent lamp is more subject to damage from direct mechanical blows than is the tungsten lamp

because of its extended tubular shape. Actually study shows that it should afford greater freedom from accidental breakage than is obtained with the tungsten fixtures now installed. Direct lighting fixtures can be shallow rectangular boxes in which the tube is protected by the outside shell of the box, by the reflector and by non shattering plastic diffusing medium covering the opening. In addition to the protection afforded by the fixture the dimensions of the fixtures (Long, shallow) are such that the natural protection offered by the strength members of the ship such as channels, etc., may be used to much better advantage than with tungsten fixtures. Much less of the fixture need project below the channels to provide even general illumination. With the many obstructions found on shipboard, it is difficult to obtain even distribution of light and afford much mechanical protection to the present fixtures. But with an extended line source of light which parallels many of the obstructions it is possible to obtain good distribution of light and at the same time practicable to allow the obstruction to provide mechanical protection for the lighting fixture.

It is neither impossible nor impracticable to design water tight fixtures for fluorescent lamps, although it seems probable on a basis of usable lumens output per fixture that the fluorescent water tight fixture would be more expensive. In some respects the design problem would be simpler than with tungsten lamps. In the tungsten lamp 98 per cent of the energy is converted into heat, which serves no useful purpose and must be radiated and conducted from the fixture. In the fluorescent fixture there is roughly 50 per cent less heat per lumen and the retention of some of this heat serves the useful purpose of keeping the lamp warm. A fluorescent lamp fixture for marine use where wide temperature changes are found should contemplate enclosing the tube so as to prevent drafts, for if heat is conducted from the tube too rapidly there is a decrease in light output. If the other features of these tubes are found by actual installation to live up to expectations steam and water tight fixtures will be designed for them.

On air conditioned units these lamps offer the advantage of reduced load on the cooling system and increased comfort for personnel. The reduced total heat produced by these lamps arises solely from the fact that to obtain equal lumens lower wattages

can be installed. The total heat produced by any electric source varies directly as the wattage regardless of the distribution of the energy emitted by the lamp for both infra-red (heat radiation) and visible radiation are eventually converted into heat which must be removed by the cooling system. However, at equal lumen intensities the heat which reaches the individual is much less with fluorescent lamps for only 50 per cent of the watts consumed by the lamp go into radiant energy while with the tungsten lamp from 85 to 95 per cent of the lamp watts are radiated. To make this point clear consider the energy distribution of a 50 watt tungsten lamp and a 20 watt fluorescent lamp which gives 15 per cent more lumens than the tungsten lamp. With tungsten lamp 1 watt goes into visible radiation (light) 43 watts into heat radiation (infra-red) 6 watts are carried away by convection and conduction. On A. C. each 20 watt fluorescent lamp with auxiliaries draws 25 watts. 2.6 watts goes into visible radiation, 7.4 watts goes into heat radiation and 15 watts are carried away by convection and conduction. Adjusted to an equal lumen basis the tungsten lamp gives out just seven times as much heat radiation as the fluorescent lamp.

Shock and vibration conditions actually encountered on shipboard reduce the expected life of a tungsten lamp regardless of operation voltage, even though the shock or vibration is not severe enough to cause failure at the time of shock or during actual vibration. With the fluorescent lamp, shock and vibration will have a very much smaller effect on the average life. While computations of light costs in both cases have been on the basis of the average laboratory life of the lamp, the percentage decrease in life of the fluorescent lamp under actual service conditions will be much less than the tungsten lamp. Thus the actual cost of light on shipboard would be appreciably greater for tungsten lamps than shown in Table I while the actual cost of fluorescent light would show very little increase over tabulated figures.

The normal voltage variations encountered on shipboard will have a smaller effect on the fluorescent than on the tungsten lamp. With a tungsten lamp a one per cent change in voltage causes very nearly a four per cent change in lumens generated by the lamp. With the fluorescent lamp a 1 per cent voltage change produces only a 1 per cent change in the lumens.

Now consider the disadvantages of fluorescent lamps both as they appear at present and as they may appear in the future.

The cost of lamps and auxiliaries are high. The cost of lamp, lampholders and auxiliary equipment, at present list prices, comes to about $4.00 per lamp. Fixtures especially designed for marine use have not appeared on the market. The price may be expected to decline with wider usage (Sales of fluorescent equipment in 1939 amounted to $10,000,000.)

Unless the Coast Guard or Navy is willing to do a modest amount of pioneering, marine fixtures will probably not appear until the more profitable demands of industry have been satisfied.

When the switch is closed with a tungsten lamp, light is forthcoming in a very small fraction of a second. At present this is not the case with fluorescent equipment suitable for marine use. On A. C. the delay in lighting of a lamp at room temperature is approximately two seconds. With D. C. operation the delay is of the order of four seconds. For some purposes this delay is not important, in other cases it constitutes a serious objection. Work is now being carried on to reduce this delay and it is probable that it will be possible to make the delay insignificant for all purposes.

When single lamps are operated on A. C. there is a stroboscopic effect due to variations in light output caused by cyclic variations in current. For the white lamp there is a 35 per cent deviation from the mean light output each cycle. This effect is so slight that it would be objectionable only in special cases. On tulamp A. C. operation the stroboscopic effect is reduced to the same value that prevails with the 25 watt tungsten lamp, a 16 per cent deviation from mean output each cycle. On D. C. operation there is no stroboscopic effect. Since the glow persistence of the phosphor producing fluorescence determines the magnitude of the deviation of light output from the mean, it is probable that as more is learned about the chemistry of phosphors that it will be possible to further reduce this stroboscopic effect.

Tulamp operation is a term applied to A. C. operation where one special ballast serves two lamps. In any installation it is not essential that the auxiliaries be mounted close to the lamp. They can be mounted any place. Some fixtures are made up with all

auxiliaries contained in the fixture so the unit may be simply connected to the line by two conductors as in tungsten practice. Where the fixture contains two lamps and auxiliaries built into the fixture tulamp operation will not differ in outside connections necessary for a single lamp fixture with self contained auxiliaries. But single lamp fixtures can easily be installed for operation in pairs using one ballast per pair.

Not only is initial cost and stroboscopic effect reduced by tulamp installations but a much better power factor is obtained. Single lamp A. C. operation entails a power-factor on 60 cycle current of from .55 to .60. By adding compensating capacitance in the circuit it can of course, be corrected. The tulamp ballast makes use of the "split phase" principal in which one lamp is ballasted by reactance only and the other lamp by reactance and capacitance in series. The lagging power-factor of the resistance branch offsets the leading power-factor of the capacitance branch resulting in an over-all power-factor of above .95. Obviously tulamp operation cannot be used on D. C.

In A. C. operation the radio interference of these lamps is a disadvantage for marine use. However, it is possible at present to reduce this interference to low levels and General Electric engineers expect within the present year (1940) to reduce it so low as to be insignificant. In D. C. operation no interference is encountered.

The installation of fluorescent lamps is not as simple as with tungsten lamps. On the other hand, it can hardly be called complicated and does not call for any skill beyond the capabilities of an electrician's mate.

In view of the trend toward a wider use of A. C. on shipboard and the many advantages of these lamps over tungsten lamps for shipboard use it seems that the future should see the general acceptance of this lamp for general lighting of vessels. Because it is especially advantageous on small vessels of the types used by the Coast Guard it seems logical that the Coast Guard should follow its development closely and be prepared to do a moderate amount of pioneering in working out the "bugs" which appear in actual service.

At the present stage of development the 18 inch and 24 inch × 1½ inch diameter tubes offer an especially desirable source of light on both A. C. and D. C. for such local lighting as chart desk, workbenches, radio operating positions, stateroom desk and berth, office desk, ordnance repair, and bulletin boards. It is hoped that the author will be permitted to make several trial installations on the *Mohawk* so that actual performance under service conditions may be observed.

NOTES.

INDEX TO NOTES.

SHIPS' CARGO-HANDLING GEAR.
—North-East Coast Institution of Engineers and Shipbuilders, February, 1941.

ENGINEERING FEATURES OF MARITIME COMMISSION PROGRAM.
—Pacific Marine Review, January, 1941.

"SEA WITCH" COMPLETES MAIDEN VOYAGE.
—Motorship and Diesel Boating, February, 1941.

RUBBER-LIKE MATERIALS.
—Product Engineering, March, 1941.

HIGH PRESSURE BOILER-FEED PUMPS.
—Power, March, 1941.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

Factors Affecting Welding Costs.
—Machinery, December, 1940.

Flow Indicator for Liquids.
—The Shipbuilder and Marine Engine Builder, January, 1941.

Causes of Diesel Engine Breakdowns.
—British Motor Ship, February, 1941.

Expensive Boring Replaced by Grouted Liners.
—Metals and Alloys, January, 1941.

SHIPS' CARGO-HANDLING GEAR.

Available types of motive power for cargo handling as well as the various forms of steam and electric winches are briefly described in this paper presented by L. T. Morton at a General Meeting of the North-East Coast Institution of Engineers and Shipbuilders held February 14, 1941. Mr. Morton is general manager of the general and electrical engineering works of Clarke, Chapman and Company, Limited.

THE DEVELOPMENT OF CARGO-HANDLING GEAR.

Since early times, ships have sailed the seas with the principal object of carrying merchandise from one country where a surplus could be produced, to another country requiring that surplus. The loading and discharging of this merchandise in early times was a leisurely matter, and manual labor was the sole means employed. Even today, when labor is very cheap, manual labor is still used to the exclusion of mechanical power. An instance of this known to many is the coaling of ships at Port Said.

The introduction early in the last century, and rapid development of the steam engine as a means of propulsion for ships did not immediately bring about any improvement in the speed of handling cargo. It was actually well past the middle of that century before the first steam winches made their appearance on ships' decks. As a guidance towards the effect of this introduction, it is interesting to note that with four men at the handles of a hand winch, or "dolly," as it was called, a load of 1 ton could only be lifted at a speed 10-12 feet per minute, whereas even the earliest steam winches could lift this same load at about 150 feet per minute. Steam power for cargo handling somewhat naturally held almost undisputed sway for the next fifty years, until the development of electric power on ships began. During this period, however, several ships were equipped with hydraulically-operated winches and cranes, which form of cargo-handling gear will be referred to later.

The conclusion of the last war saw the introduction and wide adoption of the internal-combustion engine for ship propulsion. The marine steam engineers, having progressed through the stages of the early oscillating-cylinder engines, and compound engines to the triple- and quadruple-expansion engines taking steam from the tried and proved Scotch marine boiler, had settled down to the production of a reliable unit with a coal consumption of about 1.5 pounds per S.H.P. per hour. It is true the marine steam turbine had made its appearance, but its use had, on the score of capital cost and perhaps nervousness, been limited to the propulsion of warships and fast passenger liners. The figure of 1.5 pounds of coal per S.H.P. per hour for propulsion of a cargo boat was not good enough to compete with the economical internal-combustion engine, and the marine steam engineers had to set about improving the economy of their product, which they certainly have done, figures of about 1 pound of coal being now quite common for ordinary tramp cargo steamers.

All this goes to show that tremendous thought and energy have been put into the problem of transporting cargoes from one port to another in a given time at a minimum cost. It is doubtful, however, whether the same thought and energy have been devoted to the economy of loading and discharging the cargoes. An average cargo boat is probably in port about a hundred days in the year, of which period considerably the greater part will be occupied working cargo. It is obvious, therefore, that the cost of working this cargo can have a considerable effect on the profit made by the ship.

THE CHOICE OF POWER UNITS FOR HANDLING CARGO.

In drawing up the specification for cargo-handling gear of a ship for a given service, the shipowner or shipbuilder has to decide on the motive power to be employed, whether cranes, or derricks and winches will be more suitable, and the number and disposition of the powered units. The choice of motive power lies between steam, electric, Diesel engine, hydraulic ram, and Diesel-hydraulic or electro-hydraulic rotary units. The decision between derricks and cranes almost invariably results in derricks being fitted, as giving a more flexible arrangement at cheaper cost. Cranes do, however, find favor for certain coastal trades, avoiding the necessity for stowing and rigging of derricks for short sea passages. The number and disposition of the powered units depend largely on the size of the vessel (which governs the number of hatches to be served) and to a lesser extent on the type of cargo to be handled. The important decision to be made, therefore, is that of the motive power to be used.

Steam and electric drive for cargo winches each has its advantages and disadvantages. Diesel drive, though economical in fuel consumption and comparatively cheap in first cost (provided some maker's standard mass-produced engine can be employed), lacks the flexibility of control so necessary for cargo handling. For safety and convenience a governed-speed constant-running engine must be used, driving the hoisting barrel through the necessary gearing and a friction clutch. Thus, creeping speeds necessary for breaking out cargo can only be obtained by slipping the clutch, a somewhat indefinite method, and light-hook speeds are limited to the full-load hoisting speed. Increased flexibility can be obtained by the introduction at a considerably increased cost, of a controllable hydraulic coupling between the engine and gearing. These characteristics have resulted in this type of drive being limited to lighters and certain small coasting vessels, which rely largely on shore gear for handling their cargo.

Hydraulic rams were introduced by certain owners for handling cargo before the end of last century with the sole intention of avoiding the noise which was and still is, to a lesser extent, invariably associated with the steam winch. Being very inefficient, the use of hydraulic rams on shipboard has almost completely ceased, while another serious disadvantage of this system was the liability of the water to freeze in cold climates. Rotary hydraulic-power units, in which the loss of the hydraulic medium is negligible, use oil as a medium and do not therefore suffer from this latter disadvantage. Their efficiency is not high, and though claims are made for 80 to 90 per cent for certain systems, in practice 60 to 70 per cent is nearer the mark. This means that whenever a hydraulic unit is utilized to transmit power from a prime mover—whether steam engine, Diesel engine, or electric motor—to a winch, about 50 per cent more power must be put into the prime mover for a given power at the winch barrel than would be the case if the prime mover was directly connected to the winch. In addition, of course, the cost of the installation is very considerably increased.

It may be assumed with safety that the introduction of hydraulic transmission of power in deck machinery was due to the lack of reliable and satisfactory means then available of controlling electrical power, when the advantages of electrical power were first being utilized for working cargo. Now that comparatively reliable and satisfactory control systems are available for electric motors, there is very little to justify the cost of such an installation in any average cargo vessel. There is, however, one field where hydraulic power may still be of interest. Some small modern coasting vessels are each fitted with a Diesel-driven hydraulic pump in the engine-room which can supply power to a pair of winches, or to the windlass and capstan, or to the steering gear, none of which groups is required to be in

service simultaneously. Where space is limited both on deck and in the engine-room and the engine-room personnel is small, there are advantages to be claimed for this system. On the other hand, however, owners may hesitate to put all their important auxiliaries at the mercy of one Diesel engine. The conclusion reached almost universally when considering the question of motive power to be employed for working cargo is that direct-driven steam or electric winches are the most suitable, in that about 99 per cent of ships afloat today are fitted with one or other of these types.

When Diesel-engined ships made their appearance, it became almost standard practice for these to be fitted with electric winches, while steamships continued to be fitted with steam winches. The introduction of electrical power for working cargo was largely for convenience, as no steam was available or required for other purposes, and the advantages of electricity were not then fully appreciated, while electrical power units on ships' decks were suffering from considerable teething troubles. The first essential feature of a cargo winch is reliability, and reliability under worse conditions of working than any other item of machinery on a ship. The reason for this is that the winch is handled by unskilled workmen who have no interest in the upkeep of the winch, and whose employers' payment varies proportionately with the speed at which the winches are driven. There is thus no incentive towards careful handling. Nevertheless, steam and electric winches can now stand up to the most arduous duties with very few stoppages, provided they are given the necessary routine maintenance and repair that all machinery requires.

THE ECONOMY OF CARGO HANDLING.

Assuming then that both types of winch are sufficiently reliable for their duties, there comes the question: "Which is the more economical?" To arrive at a decision there are three main points to consider, which are, the capital cost of the installation, the fuel cost to operate it, and the cost of maintaining it in good order. Electric winches cost considerably more than equivalent steam winches to buy—roughly, two and a half times as much—without taking into consideration the costs of boilers and steam pipes and generating plants and wiring; but they cost considerably less to run. It is difficult to obtain accurate figures for the relative cost of fuel for working a similar cargo in or out of similar ships by steam and electric winches, but figures have been published making it almost ten to one in favor of electric winches. Probably about seven to one is a fair figure. One factor which greatly favors electric winches is that stand-by losses are reduced to zero, while in a steam outfit radiation and leakage losses are continuing whether the winches are working or not. It is not practical to shut steam off deck for stand-by periods of short duration.

Relative maintenance costs naturally depend on the types of winches being considered, the treatment they receive, and the attention given to them in routine overhauls, but it is generally accepted that, again, electric winches show a considerable saving over steam. From the cost of running and maintenance then, there is little to recommend the fitting of steam winches, but their lower capital cost gives them a great advantage, which has resulted in the majority of ships built in this country at any rate being equipped with steam winches. It is not economy, however, to allow initial cost to be the sole guide towards the settlement of which is the more economical type of winch for working cargo. There is no doubt that many ships at present in service fitted with steam winches could have earned considerably more profit for their owners if they had been fitted with electric winches. The converse, of course, is equally true, namely, that it is not economy to equip a ship with expensive electric winches, however economical they may

be in fuel consumption when working, if they are to lie idle for long periods on the deck. It is, however, a comparatively easy matter, knowing approximately the amount of cargo to be handled in a year, to calculate whether or not the extra expenditure on electric winches can be justified.

There is no need for this calculation to be affected in any way by the method of propulsion chosen for the ship, though apparently it often is. Consider the case of a shipowner who favors steam propulsion and is building a cargo tramp ship destined for a trade with comparatively short sea passages and frequent spells of cargo working. Surely the ideal equipment is an outfit of electrically driven winches taking power from one or more Diesel-driven generators, and such engine-room auxiliaries as are required for port use arranged for electric drive. Though the Author is without experience in this matter, he feels that where and when heating is required in port it could be provided electrically as cheaply as by steam. The result of such a policy would be that boilers could be shut down in port, and the extra capital charges due to such an installation quickly recovered by the resulting savings in port fuel costs. The provision of storage capacity for a small amount of Diesel oil, in addition to that for coal or boiler oil, is hardly an important disadvantage.

RELIABILITY AND SUITABILITY OF STEAM AND ELECTRIC WINCHES.

Steam winches suffer, somewhat naturally, considering the conditions under which they work, from a fairly high rate of wear and tear. Corrosion is going on in the cylinder castings all the time unless careful attention is paid to drainage at the conclusion of each period of cargo working. The wearing parts are, however, comparatively cheap and easy to obtain and, provided the makers are a reputable firm, easy to fit. Apart from wear and tear, of course, breakages often occur, sometimes due to failure to replace worn-out parts, occasionally perhaps due to faulty design or material in the winches, or to extraneous causes. One common cause of failure is water coming through the steam pipes. No winch running at speed can take solid water through the cylinders without suffering some damage. Unfortunately, water troubles are very difficult to overcome in deck steam pipe lines, supplying steam to intermittently working winches. All that can be done is to provide efficient insulation for the pipe lines, suitable arrangements for drainage, and perhaps the addition of a certain amount of superheat to the steam, and avoid pockets in steam pipe lines for the collection of water.

Electric winches have for some considerable time been proved entirely reliable, but it would be folly to profess that such has always been the case. It is possible that failures of early electric winches are still influencing certain owners against their wider adoption today. Electric motors that had proved entirely satisfactory for land use when put to work on ships' decks proved totally inadequate to stand up to the arduous conditions imposed on them. Water found its way through joints previously considered watertight, insulation failed, armatures burst, contacts failed to stand up to the treatment of unskilled labor, and a host of troubles arose which quickly brought about the necessary improvements in design and construction required for reliability. To ensure satisfactory service from electric winches, a certain amount of routine attention from skilled personnel is essential. Given this attention, electric winches probably cause considerably less stoppages of cargo working than steam winches. Such stoppages as occur, however, though perhaps serious in their consequences, are usually very simple in themselves, being due to such troubles as blown fuses in contactor operating circuits, burnt-out relay or contactor operating coils, burnt contacts, or mechanical defects in the control gear, all of which faults can be quickly located and repaired by skilled electricians or engineers. When major electrical breakdowns do

occur, such as the burning out of an armature, they are generally due to faults in workmanship or material in the winches, as the protective features embodied in modern electric-winch control systems almost preclude such damage through careless handling.

Coming to the question of suitability, there is no doubt that the inherent characteristics of the steam engine are more suited for the duties of cargo handling than are those of the electric motor. The former is able to exert maximum torque at slow speeds for breaking out cargo, accelerates rapidly owing to low mass of rotating parts, and can sensitively control the lowering of loads. A normal series-resistance-controlled motor requires artificial load, generally provided by a foot brake to give slow speeds, and tends to give a jerky pull when accelerating as each step of resistance is cut out, and owing to the inertia of the armature, has a slower acceleration. Generally, no inherently controlled lowering of loads is provided by a series-resistance-controlled motor. This can be obtained by designing the motor field with a preponderance of shunt windings, which unfortunately reduces the speed range of the motor and hence the light speeds. An added disadvantage of a winch arranged for electrically controlled lowering, or dynamic breaking as it is technically termed, is that it is unidirectional in operation, in that the winch must be running in a specified way for hoisting. While this causes no difficulty when using a fixed rope on the center barrel, for whipping cargo or topping derricks from a warp end, care must be taken to put the ropes around the whip or warp end in the correct direction.

The most widely accepted type of winch, fitted with foot brake for lowering and light speed control, a magnetic brake for holding load automatically in the "off" position of the controller, and a centrifugal brake for limiting lowering speeds to a safe figure, does, however, give an entirely satisfactory means of handling cargo. Several types of control for winch motors have been developed which give the motors characteristics equivalent to that of a steam engine and of which mention will be made shortly; but these considerably increase the cost of an outfit of winches, and their use has been largely confined on shipboard to more vital machines such as the windlasses and capstans. Where silence while working is essential or desirable electric winches, speaking generally, are infinitely more suitable than steam winches, as anyone who has endeavored to sleep underneath or anywhere near a normal steam cargo winch at work will fully appreciate. Electric winches are always ready for instant service, whereas steam winches require warming through, and the steam pipes thoroughly clearing of water before they can do their work. Opening up the steam valves for deck steam is often a very tedious operation, in cold climates particularly, any haste causing alarming noises and broken joints in the steam lines. Engineers who have had to lie in snow trying to thaw out frozen steam pipes with blow lamps in inaccessible places on deck, will doubtless fully appreciate the availability for instant service at any time of electric winches.

It is often stated that electric winches are faster working than steam winches. The size and power of winches are specified by shipbuilders or shipowners when inquiring for quotations, but both types of winch can be supplied to lift any load at any speed within reason. For optimum results, of course, the speed of lift of cargo winches should bear some direct relation to the height of lift required. The time required for preparing a load for lifting is practically constant whatever the size of the vessel, and the speed of the winch should be such that a load can be discharged or shipped and the hook returned for another load in time to meet the completion of its preparation for lifting. Accordingly, small coasters are fitted with comparatively slow winches capable of lifting full load at about 50 F.P.M., and passenger cargo liners with high trunked hatches with winches giving speeds of 150 F.P.M. or more on full load. When considering the power or speed

of haul per winch to be installed, however, the initial cost of the winch, which varies with the power, is not always the only factor to consider. In the case of steam winches on a steam-propelled vessel, it is almost certain that sufficient steam capacity will be available on the vessel to meet any reasonable requirement. In the case of steam or electric winches on a Diesel-engine-driven vessel, however, the size of the boiler or of the generating plants respectively will be generally determined by the power of the winches, and the cost of the said boiler and generating plants must also be taken into consideration. In either case, of course, the fuel cost per hour to operate the winches will vary almost directly with the winch power installed.

TYPES OF STEAM AND ELECTRIC WINCHES.

The foregoing remarks have covered briefly the relative advantages and disadvantages of steam and electric drive for cargo winches, the comparisons being made on general lines between the types most commonly used on shipboard. There are and have been, however, many variations of both steam and electric winches available on the market, and to obtain a true picture of the efforts that have been and are being made to provide improved cargo-handling machinery it is necessary to make mention of these various types of steam and electric winches.

Steam Winches—The normal type of steam cargo winch in common use today differs but slightly in all its details from the first steam winches produced. All working parts are exposed to the atmosphere, and lubrication is provided when the donkeyman thinks fit. It must not be assumed, however, that the preservation of the normal steam cargo winch in its original form is due to the inability of winch makers to design and produce a more refined article. It is the demand for an article that creates its supply. Though comparatively rarely called for, there are refinements available for owners who are prepared to pay the extra price unfortunately but of necessity entailed, designed with the object of giving longer life to wearing parts and producing a quieter working winch. Lubrication, an important item often not given sufficient attention, is normally by oil in preference to grease. Such cover as is provided for oil cups and oil holes must be easily and instantaneously removable for the application of oil, as bearings must be lubricated without interfering with the running of the winches. It is not possible, therefore, to make oil cups and oil holes watertight, and such oil as is supplied to the bearings is generally accompanied by water and other foreign matter which does not make the already arduous duty of a winch any easier. If watertight covers were provided it is doubtful if lubrication would be supplied at all until its absence was obvious. To overcome this difficulty grease lubrication can be and has been supplied, but not on a large scale. It is definitely easier and safer to pour oil into a cup or hole than to screw down a grease cup or attach a gun to a grease nipple, especially on parts that are liable to start in motion without warning. As has been mentioned earlier, winch drivers have no interest in the upkeep of winches.

Much of the noise of a standard steam winch emanates from the spur gearing, which is normally of the moulded tooth type. A considerably quieter and more efficient winch is obtained by the fitting of machine-cut gear running in oil-tight gear cases, and the life of the gearing is prolonged almost indefinitely, but again, of course, extra initial cost is entailed. The Carron type of worm-geared steam winch is quiet and clean, all engine parts and gearing being totally enclosed and running in oil. Unlike the normal spur-geared winch with single and double gear, only one speed is obtainable on this winch and its steam consumption is high, engine control being by a differential valve which permits of no cut-off being provided.

Electric Winches—Electric winches also appear in a variety of forms, and several technical papers have been read before this and other institutions on the subject of the motors and control systems in use for electric winches and other electric auxiliaries on shipboard. The simplest and cheapest type of electric winch available on the market is undoubtedly the two-speed spur-geared drum-controlled type, which is largely fitted by Continental shipowners, but has not been adopted to any great extent by shipowners in this country. Though cheap in first cost, drum control is not generally considered suitable for the heavy duties of cargo handling, in the hands of unskilled labor. Some winches of this type do not possess the advantage of silent working generally attributed to electric winches, and are in fact, noisier than some steam winches. The type of electric winch in general use by shipowners in this country is the worm-geared contactor-controlled type, and it was on this type of winch and a standard steam winch that the cost ratio between steam and electric winches given earlier in the paper was based. A totally enclosed watertight motor mounted on the winch bedplate drives the barrel shaft through worm gearing running in an oil-tight gear case. If whipping drums are required in addition to the barrel and warp ends, these can be provided by driving a whipping shaft through a step of spur gear from the barrel shaft. The motor is controlled by a small master controller which operates the contactors. Overload and no-volt protection is provided, also relays to prevent too rapid reversal of the motor, while the contactors are interlocked for the correct sequence of operation and controlled either by current lock-outs or fixed timing devices for the correct speed of operation. To enable the full horsepower of the motor to be used on light loads, a load discriminator is fitted, which takes the place of a mechanical change speed gear, but its action is automatic. An isolating switch is fitted, so that each winch of an outfit can be made " dead " independently for overhaul or maintenance work. An electric brake is provided which releases as soon as the controller is moved away from the off position; also a centrifugal speed limiting brake, and a foot brake, all being totally enclosed and watertight. The latter brake operates a switch, which inserts resistance in the motor circuit when the brake is applied, and allows the motor to be stalled by the foot-brake without causing any damage, or bringing out the overload relay.

The position of mounting for the controller and control gear is optional, but the general rule is for the controller to be mounted on top of the motor, and the control gear and resistance either in compartments formed in the winch bedplate, or in a watertight case mounted on the bedplate, in both the cases, the necessary ventilation being provided. Alternatively, the control gear and resistance can be mounted in deck-houses, so that the necessary maintenance work can be carried out on the equipment in any weather. It is surprising that this arrangement is not more widely adopted, as it is almost certain that the slight loss of deck space caused by the provision of deck-houses would be more than compensated for by the greatly increased facilities for the proper care of and attention to the control gear. Another advantage not often claimed for winches with the control gear mounted separately, but present nevertheless, is that there is no obstruction behind the winch barrel, and whipping can conveniently be done from the barrel in addition to the warp ends and whips, if fitted. Electrical faults due to ingress of water through ventilators, accidentally left open are absent when the control gear is mounted in a deck-house. Master controllers may also be placed remote from the winch, but for satisfactory control of contactor-geared winches, the foot brake must also be operated from the position chosen for the controller. Occasionally, one finds winch controllers placed in front of their respective winches so that the operator can overlook the hatch. Winches have also been supplied with portable double controllers for each pair of winches,

connected to the winches by flexible multi-core cables, so that one operator can work cargo, looking down the hatch, or over the ship's side as required.

The foregoing remarks on the position of the controllers and control gear, though included in the description of contactor geared winches, are in general applicable to all types of electric winch. Of the special types of electric cargo winches on the market, probably the best known is the "Wilson" winch. Though considerably more expensive than an equivalent normal contactor-geared winch, its several attractive features have led to its standardization on the vessels of more than one important company. Its main advantages are short fore-and-aft dimensions enabling large hatches to be provided in a given deck space, and almost complete silence in operation (so much so that stevedores below have been heard to complain bitterly that they do not know the hook is coming until it hits them!). The winch consists of a large-diameter slow-speed motor driving a co-axial barrel through epicyclic gearing, the planet pinions of the epicyclic gear being mounted on a spring-controlled torque arm which operates a variable load discriminator. The controller has three steps hoist and lower, the first step hoist being an armature diverter step to give a slow speed for tightening slings, followed by two series resistance steps, while the lowering side has three steps of potentiometer braking. Loads are held in the off position by the motor being short circuited across itself with the shunt field applied.

Other types of electric winch employ variations of the well known Ward Leonard principle for control, such as "booster," "reducer" and "convertor" control, all of which have formed the subject of previous papers before technical institutions. Once teething troubles were overcome, the results in service of winches fitted with these types of control have supported the claims made for them of smoother operation, more delicate control, and reduced maintenance costs. Their initial cost is high, however, and as perfectly satisfactory results can be obtained from modern contactor-controlled winches, it is doubtful if their adoption can be justified on an economic basis. When considering the cost of electric winches particularly, the cost should be that of the type, not the quality. A cheap type of winch of expensive quality is, of course, infinitely preferable to an expensive type of winch of cheap quality. An early attempt to overcome the difficulty of controlling electric motors satisfactorily was the Macfarlane winch, in which a constantly running motor drove a number of barrels, generally four, through epicyclic gears engaged at will by external brake bands. The difficulty of maintaining the various brake bands in proper adjustment made this winch somewhat unreliable, and very few (if any) are in service today.

DERRICKS.

The subject of cargo-handling gear is not complete without some reference to derrick arrangements, and heavy lift derricks. Derricks were originally of timber construction, but are now almost universally of steel, tubular in form, having less weight for a given strength. The number and disposition of derricks fitted, as mentioned earlier, depends entirely on the number and size of hatches to be served. A normal tramp cargo steamer has two derricks per hatch, mounted one on either side of the ship's center-line, the derrick heels being on a table running athwartship some six feet above deck level, and the topping leads being taken from either to cross trees on the masts, or from sampson posts to the derrick heads. For cargo working, one derrick is trained over the side and the other over the center of the hatch, the load being lifted on one derrick, transferred in mid-air, and lowered by the other derrick, with each derrick guyed in position.

Small coasting vessels are generally fitted with slewing derricks, each mounted on the center-line of the ship, which work cargo in or out of the

ship by slewing inboard or outboard, controlled by man power or by small slewing winches. For slewing derricks handling heavy or difficult loads, where it is desirable to plumb any part of a hatch with the hook the topping lead is led to another winch barrel so that the angle of the derrick can be adjusted at will. The controls of this winch and those of the hoisting and slewing winches, are so grouped that one operator can handle all motions. For normal derricks, the topping lead is shackled to the deck, and adjustments made to the derrick angle when required by taking a few turns around a winch warp end and hoisting or lowering. In all cases the hoisting rope lead is taken from the winch barrel back to the derrick heel block, and through a block at the derrick head to the hook. The distance between the winch barrel and the derrick heel block is of the order of 6 to 8 feet. This short distance, therefore, does not allow the wire to spool correctly on the barrel, and it tends to pile up on the center of the barrel. To improve the working life of a wire, surely a better arrangement would be to take the hoisting lead some distance up the mast or sampson post and through an eye formed in the derrick head. With this arrangement of wire lead it would be possible to mount winches on the derrick tables, or deck-houses, and allow for longer hatches to be provided in the same deck space. Careful attention should be paid to rope guards, to prevent slack wire coming off the barrel of a winch and becoming wedged between the barrel boss and side frame.

Heavy-lift derricks form part of the cargo-handling equipment of many units, designed for lifts of up to 150 tons in some cases. A well-known fleet of ships, owned by Christen Smith, Oslo, was designed entirely for heavy-lift transportations. Heavy-lift derricks are placed of necessity on the center-line of a ship, the heel being fitted to the deck, which must be suitably stiffened locally to withstand the heavy loads imposed thereon, and the mast to which the derricking and hoisting leads are carried must be securely guyed. Heavy lifts are transported inboard or outboard by slewing the derrick by power through multi-purchase blocks. As a general rule, the winches used for heavy derrick work are quite normal winches, apart from being fitted with long barrels to accommodate large quantities of wire, and the heavy pulls for lifting the loads and topping the derrick are obtained through the necessary purchase blocks.

ENGINEERING FEATURES OF MARITIME COMMISSION PROGRAM.

Below appears an abstract of a paper written by J. E. Schmeltzer, Associate Technical Director of the Maritime Commission, read before the annual meeting of the Society of Naval Architects and Marine Engineers on November 14, 1940. The abstract is reprinted from the January, 1941, issue of Pacific Marine Review.

The power to propel a ship increases approximately as the cube of the speed; therefore, when the speed is increased above current practice, the power is also increased considerably. To make such a ship practical, it is necessary to design an economical power plant, as otherwise the increase in fuel consumption per knot would prohibit the increase in speed of the ship.

The Commission's first consideration, therefore, in the design of the main power plant for its cargo ships was to arrive at the most economical design consistent with low first cost, available materials, weight and manufacturing

facilities. This study resulted in two fundamental designs; namely, a steam plant, using cross-compound geared turbines, and a diesel plant, utilizing in one case the direct-connected Sun Doxford engine and, in the other, geared diesels.

STEAM PROPULSION.

The steam plants on the Commission's C-1, C-2 and C-3 cargo ships are fundamentally the same, except in horsepower, and consist of two water-tube boilers of either the sectional-header or two-drum types, utilizing air preheaters or economizers, or in other cases both preheaters and economizers, with water-cooled furnaces to give an overall boiler efficiency of 88 per cent at normal load with a feed temperature of 318 degrees F. The forced-draft fans are motor driven, one fan per boiler. The combustion rate is automatically controlled under all conditions of operation. All of the Commission's later plants are equipped with wide-range burners.

The boilers generate steam at 450 pounds per square inch gage pressure and 750 degrees F. total temperature at the superheater outlet, and are equipped with internal desuperheaters to supply the necessary quantity of desuperheated steam for heating purposes. The main feed pumps have been of either the motor-driven reciprocating or the turbine-driven centrifugal types.

The main turbines are of the conventional cross-compound, double-reduction geared type bleeding steam non-automatically for three stages of feed heating. The turbines exhaust to the main condenser, which in all cases is located directly under the turbine exhaust. The steam conditions for which the turbines are designed are 440 pounds per square inch gage pressure and 740 degrees F. total temperature. The astern turbine is located in the exhaust of the low-pressure cylinder and consists of one Curtis wheel to give 80 per cent of normal torque at 50 per cent of the normal ahead revolutions with 100 per cent normal ahead steam flow. The ahead turbines are of either the impulse or the combination impulse-reaction types.

The main thrust bearing is of the segmental type and is located in the gear housing. The tooth loading on the main gear is limited to 60 pounds per inch face per inch pinion pitch-circle diameter. The main turbines and gears are lubricated by a common gravity system, the sump tank being located in the gear case.

Feed heating is accomplished by two closed and one direct-contact deaerating-type heaters to give a feed temperature of 318 degrees F. with zero oxygen by the Winkler method.

The main condenser is of welded construction with cast-iron water boxes of either the two-pass or single-pass design, with a maximum water velocity of 7 feet per second. The main condenser is designed on the basis of 85 per cent clean tubes to give a vacuum of 28½ inches with the circulating water at 75 degrees F.

In selecting pumps, the Commission decided to use motor-driven centrifugals as far as possible, with steam duplex pumps as standbys in some cases. This eliminates a great amount of steam piping, flanges, valves and fittings and also many small inefficient turbines or inefficient duplex steam-driven units.

Two geared auxiliary turbo-generators supply the necessary electric power, with one acting as a standby. The turbines operate on the same steam conditions as the main plant, and exhaust into their own individual condensers at a vacuum of 28½ inches. The condensing system is designed with the same factors as the main plant. Twin two-stage air ejectors mounted on inter and after condensers are used for removing air from both

the main and auxiliary condensers, one element acting as a standby. The auxiliary turbo-generators are arranged to exhaust to the atmosphere in an emergency and to facilitate dead ship starting.

The main generators are of the 3-wire, 240/120-volt design with 25 per cent unbalance.

In the C-1 ships, all motor controls in the engine room are located in a common cubicle, permitting any motor to be started at one point. Additional pushbuttons, however, are located at the motor to permit stopping and starting locally, if desired.

Steam is generated in a contaminated evaporator at 50 pounds per square inch gage for use in the fuel-oil heating system as required.

Make-up feed is supplied by an evaporator which uses steam from the main turbines, with the evaporator supplying heating steam to the first-stage heater and draining into the deaerating heater.

All piping is properly insulated for the temperatures involved, to limit the surface temperature to 125 degrees F.

Motor-driven ventilating fans and adequate ducts insure complete and proper ventilation of the machinery spaces.

The main propeller is of the air-foil-section, variable-pitch type.

The S.S. *Challenge* was the first steam-propelled ship of the Maritime Commission design to go into service. Her performance established what is believed to be a new world record for a ship of her type and class. Her fuel rate was 0.545 pound per shaft horsepower per hour under trial conditions, which is about 0.575 pound for all purposes.

DIESEL PROPULSION.

The power plants of the motorships of the direct-connected type consist of one 4-cylinder, 2-cycle, directly reversible engine directly connected to the propeller shaft. The engines have a bore of 32 inches, a combined stroke of 95 inches and a piston speed of 728 feet per minute. The pistons and cylinders are cooled with fresh water. Auxiliary power is generated by 275-kilowatt diesel-driven generators.

Both the main and auxiliary engines operate on the same type of fuel, namely, a high grade bunker "C."

To assist atomization and burning, the fuel is heated to approximately 180 degrees F. Chromium-plated liners are also being tried, with the object of reducing liner wear. These engines are operating with jacket-water temperatures slightly higher than normal, namely, 170 degrees F., and arrangements are being made for thermal control. This is found to be quite necessary, particularly at light loads, if the exhaust is to stay clear and the engine is to operate smoothly and handle the heavier fuel.

Fresh water in conjunction with a water softener is used for cooling the engines. The cooling water is sent to a heat exchanger, with sea water as a cooling medium.

Steam for heating is generated by means of a waste-heat boiler, supplemented by an oil-fired section. All other features of the ship are essentially the same as previously described for the steamship.

The first ship of this type to go on trial showed an oil rate of 0.42 pound per shaft horsepower per hour for all purposes.

The other design of motorship developed by the Commission consists of either four or two main engines driving the propeller through reduction gearing, utilizing either hydraulic or electro-magnetic slip couplings. The main engines are of the 2-cycle type operating at approximately 240 revolutions per minute and not exceeding a piston speed of 1110 feet per minute and a brake mean effective pressure of 60 pounds per square inch at normal load.

All of the auxiliaries are the same as those installed on the direct-connected diesel motorship.

The Maritime Commission is the first to use electro-magnetic slip couplings for merchant service in this country, and those now in service give evidence that a good selection was made. They not only eliminate torsionals, but permit greater flexibility in maneuvering. The overall efficiency of a coupling is more than 97 per cent.

DECK MACHINERY.

A brief summary of improvements in the design and construction of deck machinery indicates that:

(1) Anchor handling speeds have been increased to an acceptable figure, namely, 30 feet per minute.

(2) The windlass control has been arranged to differentiate between anchor handling and warping without sacrificing protection for the equipment.

(3) Winch and capstan heads have been increased in size and whelps have been eliminated, thus increasing the life of the rope.

(4) Warping capstans have been equipped with an overhauling drive, and the control is similar to that for the warping portion of the windlass, which in effect produces adequate warping power and high light-line speed and limits the maximum line pull to a safe value.

(5) Cargo winch power and speeds have been increased in order to handle satisfactorily a high percentage of 3- and 5-ton loads. Gravity lowering has been eliminated, and the motor control equipment provides for automatic acceleration both when hoisting and lowering, as well as for dynamic lowering and automatic retardation of overhauling loads.

(6) Waterproof enclosures are being provided for the electric brakes, clutches, etc., used on deck machinery located on the weather deck.

(7) Steering-gear pumps have been improved by the increased use of anti-friction bearings. The universal use of one of the several types of "storage motion" control has practically eliminated control casualties on the electric hydraulic type of steering gear.

(8) The type of davit and lifeboat-handling equipment have been improved with respect to safety, and the power of the winches has been increased so as to hoist the lifeboat and its complement at a higher rate of speed than previously.

CARGO REFRIGERATION.

Cargo refrigeration differs strikingly from the older systems now in use in merchant service. The old system, using brine with its complications, frosting problems and heavy weight, has been replaced by direct-expansion units in each box. Usually, a separate compressor is installed for each box, giving greater flexibility and economical operation under any condition of temperature or type of cargo. Any temperature can be maintained from 0 degree F., permitting the handling of any cargo from frozen products to bananas. Frosting problems have practically been eliminated, as the suction pressure on the compressor is now more nearly equivalent to the temperature desired. For the same reason the compressors are more efficient. Should any box be empty, it is unnecessary to operate its compressor, resulting in additional saving in power consumption.

PACIFIC LINER SPECIFICATIONS.

By far the most outstanding engineering achievement thus far is the P-4-P design, which is now in specification form. This ship is to be 760 feet in

length, with a beam of 98 feet and a draft of 31 feet 9 inches. It will have a built weight of 23,000 tons and a deadweight of 18,000 tons, giving a total displacement of 41,000 tons. The deadweight will be composed of 8900 tons of fuel oil, 700 tons of fresh water, 1100 tons of cargo oil, 6000 tons of dry cargo, 600 tons of refrigerated cargo and 700 tons of miscellaneous stores.

This large ratio of deadweight to built weight is the result of many factors. First of all, the hull is welded as far as possible and it is so designed that no expansion joints are necessary. There is likewise a great saving in machinery weight, due to the layout as well as to the design. A large evaporating plant removes the necessity of carrying fresh water and, as a result, increases the deadweight 3500 tons, which is exactly the total combined weight of all the main propelling and auxiliary machinery. This great reduction in weight provides an opportunity for a better arrangement of the fuel-oil and ballast tanks, giving added stability with a full three-compartment standard. Since all of the fresh water is made, there is no consequent loss in deadweight from that source and, because of the increased efficiency of the main propelling plant, there is still less loss in deadweight from the burning of fuel, making it possible in this particular design to make the complete voyage without utilizing ballast tanks.

The main propelling machinery as proposed for the P-4-P design is located in two engine rooms, each being a complete and distinct power plant and entirely separate from the other. The machinery in the after engine room will drive the starboard propeller, while the machinery in the forward engine room will drive the port propeller by means of triple-cylinder turbines and double-reduction gears.

The two main engine rooms will be separated by two smaller auxiliary rooms in which will be located the main evaporating plant. This arrangement lends itself readily to safety and stability.

Each main turbine will be designed for a normal output of 29,000 shaft horsepower and a maximum of 44,000 shaft horsepower, operating on a reheat regenerative cycle in contrast to the Rankine cycle, which is now common practice. Steam will leave the superheater outlet at 1200 pounds per square inch gage and 750 degrees F. total temperature. The steam will exhaust from the high-pressure turbine at approximately 300 pounds per square inch gage. It will return to the reheater section of the main steam generator and there be reheated to 750 degrees F. total temperature before entering the intermediate-pressure turbine.

The steam generators will be equipped with economizers and air preheaters, giving an overall efficiency of approximately 88 per cent. The rate of combustion will be controlled automatically, thereby maintaining efficient operation at all loads. Compressed air will be used in place of steam in the soot blowers, thereby eliminating a large amount of steam piping to the soot blowers and at the same time reducing the amount of make-up water to the system.

The main condenser will be of the single-pass design, the circulating water being supplied by scoops and thus eliminating the use of large main circulating pumps. The feedwater will be heated to 400 degrees F. by means of three closed heaters and one deaerating heater.

There will be four 1000-kilowatt turbo-generators operating on steam at 1200 pounds per square inch and 750 degrees F. at the throttle, bleeding at 50 pounds gage non-automatically for heating and automatically at 10 pounds gage pressure for the evaporating plant and deaerating heater. An automatic bleeder was selected for this steam pressure because both the deaerating heater and the evaporator are very sensitive to changes in pressure.

The generators will be 440-volt, 3-phase, 60-cycle machines, and all auxiliaries will be motor driven, with the exception of the main feed pump, which

will be turbine driven, exhausting at 10 pounds pressure into the common line between the deaerating heater and evaporator.

All valves in the various piping systems will be remotely operated from the central control station by means of air. Likewise, all controls for vital pumps, watertight doors, fire-screen doors, zone-mechanical ventilation and CO_2 fire-fighting equipment will be located in the central control room, which will be adjacent to the bridge. This permits the operator in the central control station to have complete control of the ship in any type of emergency, whether it be fire, collision or grounding. This installation is a distinctly progressive step; it results not only in better and safer operation, but in a reduction of piping weight.

The main steam piping will be led direct from its own boiler to a common manifold incorporated in the maneuvering valve of the main turbine, thereby eliminating certain flanges and connections which otherwise would be required, if they were joined in a common header. This, likewise, permits the use of smaller piping, giving greater flexibility to imposed stresses due to temperature. Because of the higher pressure and piping arrangement, it is not contemplated using any piping larger than 6 inches in diameter.

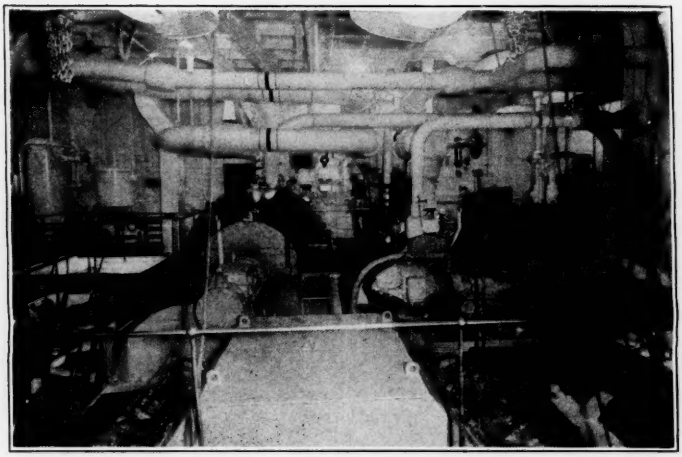
The reheat cycle has been adopted for a dual purpose. First, it reduces the fuel consumption approximately 15 per cent below that which would be obtained if a Rankine cycle were used, both having the same limiting temperature. Second, it gives control over the moisture content in the exhaust, thereby giving a flatter fuel per shaft horsepower curve. In other words, approximately the same economy is obtained at half power, normal power and full power.

By limiting the temperature to 750 degrees F., carbon steel may be used throughout, reducing the plant cost materially.

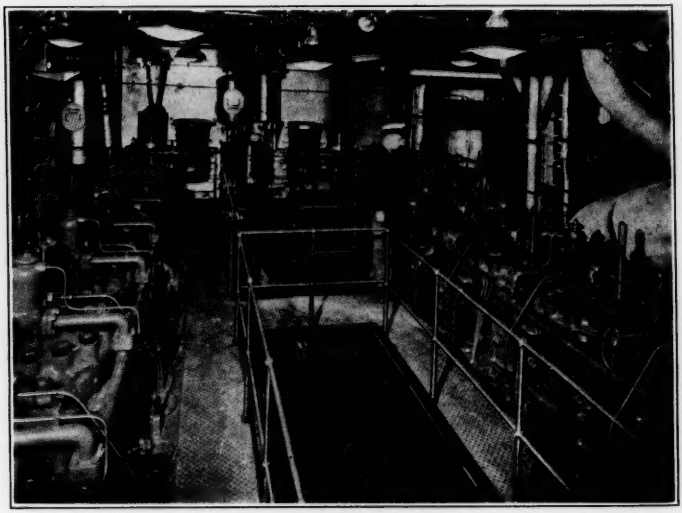
With such a plant, it is expected that an oil rate of 0.50 pound per shaft horsepower per hour for all purposes will be obtained. This includes the hotel load and the making of 64,000 gallons of fresh water per day. The rate of the *President Coolidge* is 0.72 pound per shaft horsepower per hour, and the calculated rate of the *America* is 0.634 pound. Both of these vessels carry fresh water.

Lighting on the P-4-P will be completely fluorescent. This type of lighting is particularly suitable for marine work in that it is far more efficient than Mazda lighting and at the same time gives longer life and a more artistic lighting effect. While the first installation costs more than Mazda, there is a considerable saving in the generator capacity because of the reduction in the lighting load and at the same time a considerable reduction in the air-conditioning load, which in the case of the P-4-P design amounts to 17 tons.

The most modern type of air-conditioning will be employed, each cabin and tourist-class stateroom being supplied with one or more units which will supply heating in the winter and cooling and dehumidification in the summer. Each unit will be complete with coils for chilled or hot water, centrifugal recirculating fan, air filter, drain pan and controls. Heating and cooling will be regulated primarily by thermostats which control the water flow through the coils. In addition, fan adjustment will be provided so that the volume of circulated air may be reduced for winter operation, or to suit individual desires. Hot or chilled water from the refrigerating machinery room will be supplied to the units. The units will be designed for recirculation in order to provide adequate ventilation and comfortable air movement, particularly during the summer months. About 30 to 50 cubic feet per minute per person, or 20 per cent of the total volume handled, will be outside air. The capacity of the units will be sufficient to effect a drop of 10 degrees F. in the room temperature with a relative humidity



TYPICAL ENGINE ROOM OF C-2 CARGO STEAMER. SINGLE SCREW DRIVEN THROUGH DOUBLE-REDUCTION GEARS BY CROSS-COMPOUND STEAM TURBINE.



TYPICAL ENGINE ROOM OF C-2 CARGO MOTORSHIP. SINGLE SCREW DRIVEN THROUGH SINGLE REDUCTION GEARING AND ELECTRO-DYNAMIC COUPLINGS BY TWO DIESEL ENGINES.

within the comfort zone. Such a system is neat in appearance and flexible enough in operation to satisfy individual demands, and it is reliable and economical.

The main propellers will have variable pitch with an airfoil blade section. They will be constructed of solid manganese bronze and statically balanced. These propellers will be the the largest ever manufactured in this country, and will offer quite a casting problem, requiring approximately fourteen days to cool.

Not only will the P-4-P ship be an efficient passenger liner during times of peace, but she will, likewise, be suitable for quick conversion for use as an aircraft carrier. The smokestacks are located on the side of the ship to simplify conversion during a national emergency.

EXPERIMENTAL HIGH-PRESSURE CARGO SHIP.

The Commission's cargo ships must be capable of maintaining high sustained sea speed, but, as the power increases, it must have increased economy. To obtain first-hand information, the Commission is building a ship of 8000 shaft horsepower, having a main power plant utilizing the same fundamental design as that on the P-4-P. From a cost, weight and performance standpoint, studies indicate that such a plant is practical at 8000 shaft horsepower, becomes justified at approximately 10,000 shaft horsepower and is noticeably more economical at 15,000 shaft horsepower and above. This is the first installation using reheat.

In detail, the design of the experimental ship will be as follows: Two steam generators will each supply 32,000 pounds of superheated steam per hour for normal operation and about 35,200 pounds per hour for maximum operation, at 1235 pounds per square inch gage pressure and 750 degrees F. at the superheater outlet. Approximately 26,500 pounds per hour of exhaust steam at 240 pounds gage from the high-pressure turbine will be reheated to 750 degrees F. in each unit.

Each steam generator will be of the double-casing, single-uptake, three-drum A-type with an air heater, economizer and convection superheater and reheater. A desuperheater with a capacity of 3000 pounds per hour is to be provided in the steam drum to supply steam for the auxiliaries. Water walls are to be provided for the reheater section. In order to assure good circulation, external downcomers will be provided, all of the tubes inside of the boiler casing being upcomers. Two burners of the wide-range type will be provided for the main section, and two additional burners will provide separate firing for the reheater section.

Since the hot gases from the reheater section will sweep across the boiler through both sections of tubes, it is necessary to arrange the superheater surface so as to prevent excessive steam temperatures while maneuvering or backing, when the reheater section is shut down. This is done by placing a primary superheater in the tube section on the reheater side of the unit and by placing a secondary superheater in the other section of tubes on the uptake side of the casing. The reheater section will be located below the primary superheater in the same section of tubes.

In order to eliminate solids in the steam, which might be damaging to the turbine blading, a special baffle separator is to be placed in the steam drum, which is guaranteed to prevent a carryover in excess of two parts per million of suspended solids (as determined by test) when the water has 50 grains of total solids (chlorine maximum of 30 and pH value of 10) and the water level in the drum is 3 inches above normal.

Automatic combustion control will be provided. The system is to be actuated through master controls which control the oil and air supply (dampers) simultaneously and by secondary controls which will actuate the

air dampers to adjust the fuel-air ratio, if the action of the master controls disturbs the correct ratio. For the high-pressure steam, the regulation will emanate from the steam pressure at the superheater outlet, while the reheater regulation will be actuated by the steam temperature at the reheater outlet. Manual control will also be provided for the boilers. This will be designed so as not to interfere with the adjustment of the automatic control system. To protect the reheater section, an automatic safety shut-off control will be arranged to stop the flow of oil to the reheat burners when the flow of steam falls below a predetermined rate. This shut-off control will have a manually operated trip and reset lever. Whenever the reheater burners are shut down it will be necessary to reset the solenoid control valve manually before bringing the burners back into service.

Forced draft will be supplied by two blowers—one for each boiler. The blowers are to be cross connected so that in an emergency one blower can handle both boilers at reduced load. The static pressure will be about 7 inches of water for normal operation and 12½ inches for maximum operation. To maintain efficient operation, electric smoke indicators and soot blowers of the air-puff type (as previously mentioned) will be furnished. An efficiency of 87 per cent is guaranteed for the boilers, but there is every indication from experience in land practice and from the estimates that this figure will be easily exceeded.

The main turbines are of the double-reduction, three-cylinder type, with the reversing turbine incorporated in the low-pressure casing. For normal operation the high-pressure turbine will take steam at the throttle at about 1200 pounds per square inch gage and 740 degrees F. and exhaust at about 260 pounds absolute to the reheater; the intermediate-pressure turbine will be supplied with steam at 230 pounds absolute and 740 degrees F. at the inlet and will exhaust at 40 pounds absolute, and the single-flow low-pressure turbine will expand the steam from about 37.5 pounds absolute to about 28½ inches of vacuum. The astern turbine will operate at the full 1200-pound gage pressure and 740 degrees F. total temperature.

The high-pressure turbine will be of the impulse type with five stages; the impulse-reaction intermediate-pressure turbine will have a single impulse stage followed by twenty rows of reaction blading, and the single-flow reaction low-pressure turbine will have sixteen rows of blading.

For emergency operations the low-pressure turbine can be operated alone, or, if the low-pressure turbine is damaged, the high-pressure and intermediate-pressure cylinders can operate as a compound unit with the exhaust going to the main condenser. When going astern, the astern turbine will develop about 80 per cent of the normal ahead torque at about 50 per cent of the normal revolutions per minute while being supplied with 100 per cent normal ahead steam.

To allow for expansion when the turbines are heated up and cooled off, they will be supported at the forward end by a flexible web and will be bolted directly to the gear case at the after end.

Extraction or bleeding will be provided at five points in the turbines for the various auxiliaries—at the high-pressure exhaust for the fourth-stage feed heater; at the intermediate pressure inlet for the generator turbines; at 115 pounds absolute for the evaporator and third-stage feed heater; at the low-pressure inlet for the second-stage (direct-contact deaerating heater) feed heater; and at 9 pounds absolute for the first-stage feed heater.

The turbines will drive the propeller shaft through a double-reduction double-helical gear unit having an integral thrust bearing of the pivoted segmental shoe type mounted at the forward end of the gear casing, and with a motor-driven turning gear mounted on an extension of the high-pressure turbine pinion shaft. The high-pressure and intermediate pressure

pinions will drive a common first-reduction gear, and the low-pressure pinion an independent first-reduction gear. Both of the first-reduction gears will be placed forward of the main or slow-speed gear. The low-speed pinions will be driven through quill shafts. For normal operation the various shaft speeds will be as follows: High-pressure turbine, 8910 revolutions per minute; intermediate-pressure turbine, 5000 revolutions per minute; low-pressure turbine, 4200 revolutions per minute; and the low-speed gear, 96 revolutions per minute. Forced lubrication will be supplied to the gears with spray nozzles fitted for the ahead and astern faces of the high-pressure, high- and low-speed pinions. A single-pass condenser will be located athwartship under the low-pressure turbine casing and bolted directly to the casing exhaust flange. The condenser will have approximately 6000 square feet of cooling surface. It will be equipped with the usual twin two-stage ejector, and will be arranged to handle exhaust from the high-pressure and intermediate-pressure turbines, if the low-pressure turbine should be disabled for any reason.

The condensate will be pumped from the condenser by the condensate pump and go through a combined main air ejector inter and after cooler, gland exhauster condenser, drain cooler and first-stage heater; and then to the direct-contact deaerating (second-stage) heater. At this point the system will be completely deaerated. The usual specifications call for a maximum of 0.03 cubic centimeter of free oxygen per liter. The feed pump will take suction from the deaerating heater and deliver the feed to the boiler via the third- and fourth-stage heaters and economizers.

Three feed pumps will be used. Two of these pumps will be geared, motor-driven, variable-stroke, inverted triplex pumps. Each of these pumps will have sufficient capacity to serve one boiler. In addition, and as a standby, there will be a turbine-driven centrifugal feed pump with sufficient capacity to feed both boilers. The deaerating heater will be maintained at 10 pounds gage pressure by means of a regulating valve on the steam supply.

This electrical load will be carried by two 350-kilowatt, 240/120 volt, three-wire, 1200 revolutions per minute generators.

There are some disadvantages for a power plant such as has been described in the foregoing, but the advantages certainly outweigh them. The major disadvantage is in the run of reheat piping and the addition of a reheat section to the boilers, but these difficulties are minimized by a compact machinery arrangement which places the boilers in the same compartment and just forward of the turbines. This results in very short runs of piping. Since the temperature has been kept down to 750 degrees F., it is not necessary to use special alloy steel piping, so that ordinary carbon steel will be used, as is customary for power plants using steam at 450 pounds gage. The use of the 1200-pound pressure will mean that the piping will have to be thicker but much lighter, because of the smaller sizes. On a ship of the same power and using 450-pound steam, the high-pressure steam main would be approximately 6 inches in diameter, while for this 1200-pound plant the size will be approximately 3 inches. It is easy to see that in a plant like this, where the piping is small and short in length, and where all units are compact and close together, there is a considerable saving not only in weight but in lagging, heat, and consequently in engine-room ventilation.

For this power plant the closed turbine water rate will be approximately 5.70 pounds per shaft horsepower per hour for normal operation. It is anticipated that a fuel rate of 0.50 pound per shaft horsepower per hour for all purposes will be obtained. By comparison, this plant, operating at 1200 pounds gage and 750 degrees F., is practically equivalent to a plant operating at 1200 pounds gage and 900 degrees F. without the reheat cycle.

SEA WITCH COMPLETES MAIDEN VOYAGE.

The *Sea Witch* was one of the first group of ships contracted for by the Maritime Commission, the first C-2 type cargo motorship completed with multiple engine and reduction gear drive, the first Maritime Commission ship equipped with hydraulic couplings, and the first ship completed by the rejuvenated Tampa Shipbuilding and Drydock Company. The following extract of an article which appeared in the February, 1941 issue of Motorship and Diesel Boating outlines various minor troubles experienced during a five month, 31,000 mile maiden voyage. Sister ships of the *Sea Witch* have been acquired by the Navy for conversion to Naval auxiliaries.

Seldom has a new ship been put to a more gruelling test, and come through it more successfully than the motorship *Sea Witch*, which vessel arrived in New York on December 30, 1940, after a round trip of 31,000 miles to the Far East, and left again on January 11 on a second similar voyage.

The *Sea Witch* was completed at the yard of the Tampa Shipbuilding Company in July 1940. She is a 459-foot cargo vessel of the C-2 type, built to the order of the U. S. Maritime Commission and turned over to the United States Lines for operation in the American Pioneer Line service to the Far East. She is propelled by two Nordberg Diesels of 3000 HP. each, which drive a single propeller through American Blower Company hydraulic couplings and Falk reduction gears. Auxiliary power is furnished by two Superior Diesels, each driving a 300-Kw. Westinghouse generator.

The vessel had her builders trials on July 1, 1940, followed some time later by the official acceptance trials. After the last-minute finishing touches at the shipyard were completed, the vessel left Tampa on July 31, bound for Baltimore where she was to receive her first cargo.

From that time on the life of the *Sea Witch* became a hectic one, marked by runs from port to port, with stops only long enough for the busy deck winches to hoist cargo out and in, this routine being maintained for five months and the routes covered before the ship reached New York on her

MAIDEN VOYAGE MOTORSHIP SEA WITCH

From	*To*	*Dates*	*Miles Dist.*	*Speed*	*Fuel per day, bbls.*	*Total Fuel Burned*	*per bbl. Knots*
Tampa	Balto.	July 31-Aug. 4	1166	14.57	106	376	3.1
Balto.	Phila.	Aug. 8-Aug. 9	139	15.2	126	51	2.74
Phila.	New York	Aug. 10-Aug. 11	130	15.2	148	55	2.36
New York	Norfolk	Aug. 21-Aug. 22	246	15.6	178.8	122	2.01
Norfolk	Newport News	Aug. 22-Aug. 23			No data		
Newport News	Savannah	Aug. 23-Aug. 25	453	15.6	178.8	173	2.62
Savannah	Cristobal	Aug. 27-Sept. 1	1555	15.3	153.3	638	2.36
Cristobal	Balboa	Sept. 1-Sept. 2			Canal transit		
Balboa	San Pedro	Sept. 2-Sept. 10	2940	15.5	145.7	1,157	2.54
San Pedro	Manila	Sept. 12-Oct. 3	6978	14.8	151	2,967	2.37
Manila	Shanghai	Oct. 5-Oct. 9	1247	15.03	153.9	531	2.34
Shanghai	Hong Kong	Oct. 11-Oct. 14	809	16.31	171.1	349	2.3
Hong Kong	Manila	Oct. 16-Oct. 18	630	14.7	112.9	203	3.13
Manila	Masinloc	Oct. 18-Oct. 19	118	15.8	135.1	42	2.8
Masinloc	Thursday Island	Oct. 22-Oct. 29	2271	15.36	138.9	856	2.65
Thursday Island	Brisbane	Oct. 29-Nov. 2	1263	15.47	142.6	485	2.6
Brisbane	Sydney	Nov. 6-Nov. 8	478	14.7	155.3	210	2.47
Sydney	Balboa	Nov. 13-Dec. 7	7798	13.7	153.3	3,641	2.14
Balboa	Cristobal	Dec. 7-Dec. 7			Canal transit		
Cristobal	Boston	Dec. 7-Dec. 14	2146	13.9	144.1	928	2.31
Boston	Norfolk	Dec. 19-Dec. 21	544	14.1	118	189	2.87
Norfolk	Phila.	Dec. 21-Dec. 22	140	14.0	114.2	48	2.91
Phila.	New York	Dec. 27-Dec. 30	129	15.3	117.1	41	3.14
		TOTAL	31,180			13,082	

Average knots 15.007 — Knots per bbl. 2.38 — Bbl. per mile .42.

return, totaling a distance of 31,180 miles. During this time the *Sea Witch* was strictly "on her own," it being the policy of the American Pioneer Line to make no voyage repairs elsewhere than in New York. The accompanying tabulation extracted from the ship's log, presents a comprehensive picture of the ship's activities during this period.

While the tabular data tell the story of the performance of the *Sea Witch*, it is of interest to record here some of the highlights of the voyage, as an indication of why the marine engineer's life at sea is never monotonous.

During the trip from Tampa to Baltimore noise developed in the reduction gears. Upon arrival in port the casing was removed and it was found that several articles left by workmen during installation, including a pocket knife, had been chewed up in the gears. Fortunately no harm was done to the gear teeth.

After leaving Savannah, Ga., bound for the Canal, several stops were made necessary by water in the fuel oil. This water trouble persisted and the question of how water was getting into the fuel tanks assumed the proportions of a mystery. It finally developed that the trouble was caused by an error in labeling valves by the shipyard, resulting in the steam smothering lines to the fuel tanks being opened, permitting steam to blow into the fuel.

After this error was discovered no further trouble was experienced with fuel until additional fuel was taken on at San Pedro, Calif. The fuel that had been received on board before leaving the shipyard was a low gravity Diesel fuel, but when the bunkers were filled at San Pedro the oil received was similar in character to Bunker C fuel.

On the 21-day run from San Pedro to Manila it was necessary to stop one engine at a time for a total of 11 hours to clean the nozzles of the injection valves. Incidentally, the carbon formation on these valves was of a very interesting character. It built up on the convex nozzle tip in a lump with radial projections extending out from the central mass in line with the radial fuel jets from the nozzle, having much the appearance of a miniature radial airplane motor.

Chief Engineer Cox finally eliminated this annoyance by stopping one engine at a time and advancing the fuel valve cams to give approximately two degrees earlier opening of the valves. After this was done the log records only one stop for cleaning valves, this being on the 24-day run from Sydney, Australia, to Balboa.

Incidentally, the stops made had little effect on the ship's average speed, due to the flexibility of the two-engine, reduction gear, hydraulic coupling drive. Either engine could be cut out at any time by emptying its coupling and proceeding with the other engine. This procedure proved very useful on occasions such as when the ship arrived at Hong Kong, China. The arrival at the entrance to the port on October 14 was after the port had been closed for the night and could not be entered. One engine was disconnected and shut down and the ship cruised back and forth all night on one engine, entering the harbor the next morning.

Reference to the accompanying table will show that a striking feature of the trip is the small amount of time spent at anchor. The longest time spent in any port after leaving New York was at Sydney, Australia, where five days were occupied handling cargo. This, added to the fact that the engine room crew is a small one for a vessel containing as much equipment as the *Sea Witch*, made the five-months voyage a very severe test of a vessel just out of the builder's yard.

There were a number of things that showed up that indicated the desirability of minor changes, such as insufficient capacity of auxiliary condenser, lack of protection for pneumercator pipes where they pass through the decks, inadequate protection of valves and piping against corrosion, and

similar items characteristic of new ships, but nothing that interfered seriously with the operation of the ship.

The only difficulty experienced with the propelling engines was the breaking of piston rings which occurred during the return trip, on the long leg between Sydney and Balboa. Steps have been taken by the engine builder to make such corrections as are found necessary. Otherwise the engines performed without incident as shown by the log.

Worthy of note is the performance of the two auxiliary Diesels driving the generators which furnish all auxiliary power. These engines operated steadily throughout the five months period covered by the voyage without the slightest trouble.

These generator engines operated throughout the voyage with the same low grade fuel as used in the main engines. Upon arrival in New York the engines were opened up by the builder's workmen, the first time the cylinders had been opened since the engines left the plant of the builders. All were found in good order with the exception of some carbon in the ring slots and some rings that were worn enough to warrant renewal.

Returning to the tabulation of extracts from the log we can obtain some interesting average results. It should be noted that the speeds recorded from port to port do not represent maximum speeds, because all speed reductions due to engine adjustments, fog, etc., are included and the figures shown represent average speed for the whole distance in each case. Considering the fact, however, that the thing of paramount importance in the case of a cargo ship is the total time required to transport cargo from port to port, the average speed is what counts.

Having this fact in mind, we may analyze the recorded figures as follows: The total distance traveled is 31,180 miles, neglecting the two transits of the Canal, and the average speed is 15.007 knots.

In considering fuel consumption it should be mentioned that the *Sea Witch* is provided with displacement meters and pneumercator gauges for measuring the fuel, as well as the usual sounding pipes in the tanks. The figures recorded in the log represent in each case the average of measurements taken by these three methods.

The figures tabulated herewith show the knots per barrel of fuel for the whole voyage, again neglecting the two Canal transits for which mileage is not recorded, to be 2.38. Expressing this in another way the average consumption for the voyage is .42 barrels per mile for all purposes. This is an excellent performance for the initial voyage but it will undoubtedly be bettered when the ship settles down to normal running.

A matter to be considered in this connection is that during this voyage the mean effective pressures and engine revolutions indicated that the present propeller is not entirely satisfactory. It is probable that the installation of a more suitably dimensioned propeller will improve both the speed and fuel consumption, although they are very good now.

RUBBER-LIKE MATERIALS.

As far back as 1860 Williams isolated a constituent from rubber called isoprene, and not long afterwards Tilden found that isoprene could be made into a rubber-like material either by acid treatment or simple aging. Tilden also was able to produce isoprene from materials other than rubber, so his work could be considered the basis of the present so-called synthetic rubber industry. While the Germans manufactured several thousand tons of poor quality synthetic rubber during the last war, it is in the last

decade that real advancement has been made in the industry. In 1939 about 200,000 tons of synthetic rubber was produced by the United States, Great Britain, Russia and Germany.

The following article and tables by J. Delmonte are reprinted from the March, 1941, issue of Product Engineering.

Recent threats of a shortage of natural rubber have aroused public interest in the development of synthetic rubbers. Much publicity has been given to these materials. Last spring it was announced that the Standard Oil Company of New Jersey had acquired the U. S. patent rights to the Buna type synthetic rubbers developed in Germany and used there to replace natural rubber in many applications including tires. The same company in May, 1940, announced another synthetic rubber of petroleum origin called Butyl rubber which was developed in its own laboratories entirely independent of the German Buna developments. It is possible that the latter will be lower in cost than Buna rubbers because of the lower cost of the interpolymer. B. F. Goodrich has developed a new synthetic, Ameripol, intended for tire manufacture as well as other applications. The Goodyear Tire & Rubber Company has developed a synthetic rubber, Chemigum, derived from petroleum; it is claimed that tires made of this chemical rubber give superior performance to those made of German Buna and equal to those made of natural rubber.

To the lay public these developments have been of primary interest because of the possibility of getting the cost of synthetic rubbers low enough to make them a factor in the largest market for rubber—the manufacture of tires. However, engineers have been using synthetic rubbers of various types since the announcement some years ago of one of the first in this country, Duprene, now Neoprene. Up to the present and until the effect of increased production makes itself felt, the field of synthetic rubbers in the United States lies primarily in those applications where cost of material is only a minor factor, and where their ability to resist oils, chemicals, heat, aging and oxidation combined with the moldability and flexibility of rubber is important. This combination of properties has made possible applications where leather, fabric, metal or rubber are not satisfactory and has opened up an entire field of products.

In a certain sense, the term "synthetic rubber" as applied to many materials is a misnomer, because the chemical and molecular structure of naturally occuring rubber has not been successfully synthesized. While efforts were at first directed toward exact duplication of the rubber molecule, these were abandoned when it was realized that material resembling natural rubber in chemical structure was superior in certain properties.

Some of the synthetic rubber-like materials differ from rubber in that they are thermoplastics, that is, they assume a new shape on application of heat and pressure. Thiokol, Koroseal, and Resistoflex are of this type. Neoprene and the Buna materials resemble rubber in that they are thermosetting but they do not devulcanize when processed by the methods commonly used for treating vulcanized rubber. Butyl rubber is a new approach to the problem, with a molecular structure almost fully saturated, yet it is vulcanizable.

In Table I a tabulation of the better known synthetic rubber-like materials is given, together with a short description of their origin and the forms available. In analyzing those qualities of synthetic rubber-like materials which make them particularly useful for design purposes, one will find considerable variation among the mechanical, electrical and chemical characteristics. These properties will be taken up one by one and the materials compared.

TABLE I.—TYPES OF RUBBER-LIKE MATERIALS.

TRADE NAMES	ORIGIN	INDUSTRIAL FORMS
Perbunan (Buna N) Perbunan Extra (Buna NN)	Co-polymer of butadiene and acrylic nitrile	Sheets, molded and miscellaneous. Best oil resistance of Buna rubbers
Buna S	Co-polymer of butadiene and styrene	Sheets, molded and miscellaneous. Best physical properties of Buna rubbers
Neoprene	Polymer of chloroprene	Latex, sheets, molding compounds, miscellaneous coatings, gaskets, cement, molded goods
Vistanex (Oppanol)	Polymer of isobutylene	Included in rubber mixes to improve aging. Molded forms
Resistoflex	Hydrolizing of polyvinyl acetate — polyvinyl alcohol	Tubings and sheets
Thiokol, Perduren	Condensation hydrocarbon dihalides and polysulphides polyalkylene sulphides	Sheets, powder for molding compounds and miscellaneous coatings, gaskets, cements
Koroseal, Flamenol, Knightware, Mipolam, Korolac	Plasticized polyvinyl chloride	Extruded tubes, sheets, tapes, molded forms and surface coatings
Irvolite tubing, Netco tape	Polyvinyl co-polymer of chloride-acetate	Electrical insulation and conduit
Butyl rubber	Co-polymer of simple olefins and small amount of dialefin	Tires, sheeting, rubber substitute

Mechanical: It is not an easy matter to specify and to tabulate the mechanical properties of synthetic rubber-like materials because of the infinite number of compositions which may be formulated to meet a variety of purposes. Through proper compounding it is possible to get rubber-like products which have the widest imaginable variation in physical properties, but those properties, either high or low, can be exactly obtained to suit a particular need. However, it is possible to make general comparisons with natural rubber, providing these variations are recognized. It has been suggested that a material exhibit at least 200 per cent elongation before being classified as rubber-like. While some materials will show this elongation in a rapid, elastic change of dimensions under load, others display some time-dependent characteristics before attaining this deformation. The time for attaining this elongation throws some light upon the elastic and plastic properties of the material. A typical example is shown in Figure 1 where a synthetic rubber is compared with plasticized polyvinyl butyral, which has attained considerable prominence as the plastic layer in safety glass. The materials are subjected to a constant load and deformation noted at various time intervals. Creep will continue at a slow rate as long as the load is applied. The ratio of true elastic deformation to plastic deformation for a given elongation is much greater for the synthetic rubber than for the polyvinyl resin, the deformation of which was more a plastic phenomenon.

As would be expected, those materials resembling natural rubber most closely in molecular structure, such as the butadiene and chloroprene rubbers, possess similar physical properties though this is not a rule. Grades

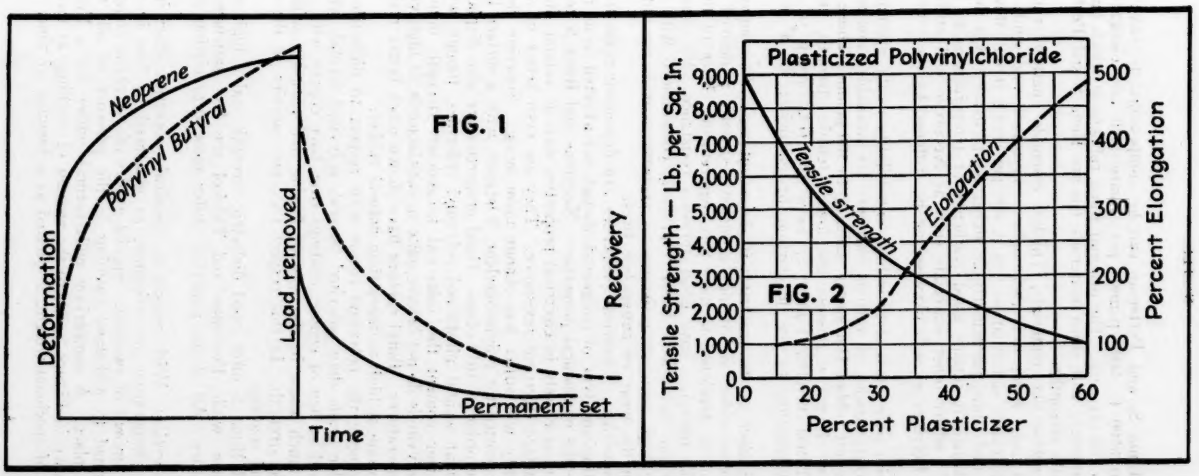


Fig. 1—Typical time-deformation properties of two rubber-like materials when subjected to a fixed load and constant temperature.
Fig. 2—Properties can be varied in compounding. Curves show effect of change in amount of plasticizer in a polyvinyl chloride

of Neoprene, Buna S, and Perbunan can be compounded to have tensile strengths up to 3500 to 5000 pounds per square inch and elongations at break of 600 to 700 per cent. The original Buna rubbers in which butadiene was not copolymerized with other materials (as are Buna S and Perbunan), had lower tensile strengths.

Among the rubber-like materials, Thiokol resembles natural rubber in appearance and feel. Since it is thermoplastic in nature, and while it will have appreciable elastic deformation (up to 600 per cent), it will flow more than Neoprene and Buna, particularly at higher temperatures. On the other hand, like Perbunan it retains good resiliency at temperatures as low as —40 degrees F., where other materials such as Neoprene will stiffen and become hard. Polybutenes will likewise not be stiffened by these low temperatures.

Polyvinyl resins owe much of their high extensibility to the presence of a suitable plasticizer. Grades of various hardness are available ranging from 20 points of the soft rubber Durometer scale, up to 100 points. Those grades with the highest elongation possess the lowest tensile strength, and vice versa. Typical test results, Figure 2, show the relationship of tensile strength and elongation to the proportion of plasticizer in a polyvinyl chloride.

Abrasion resistance of the materials is dependent upon the compounding and the ingredients added, and if high abrasion resistance is desired special compositions can be developed to meet this requirement. Most of the synthetic rubber-like materials have good abrasion resistance. By compounding this quality can be varied. Buna S properly compounded with carbon black has quite high abrasion resistance, a property that is an important consideration in the wear of automobile tires.

Electrical Properties: Electrical properties are dependent upon the particular formulation and type of compound, though not affected to as marked a degree as are the mechanical properties. Neoprene and Buna S resemble natural rubber most closely in electrical properties such as volume dielectric strength and surface electrical resistance. They are even better in one respect, in that they will suffer less change upon aging. Superior electrical properties are reported for polyisobutylene, Vistanex, with a surface leakage resistance of the order of 10^{16} ohms. Good properties are also displayed by the high molecular weight plasticized polyvinyl chlorides, though with increasing plasticizer content the volts/mil breakdown strength diminishes. However, the polyvinyls do not behave quite so satisfactorily at high voltages and at high temperatures. Butyl rubbers have shown much better resistance to corona and ozone at high voltages than natural rubber.

Outstanding among the polyvinyl resins with respect to dielectric properties is a harder grade than polyvinyl butyral, polyvinyl formal, which is being widely used in lieu of enamel coatings over bare copper wire. They will withstand much abuse in the way of flexing and abrasion with little loss in dielectric strength. In this respect they are superior to the earlier developed enamel coatings.

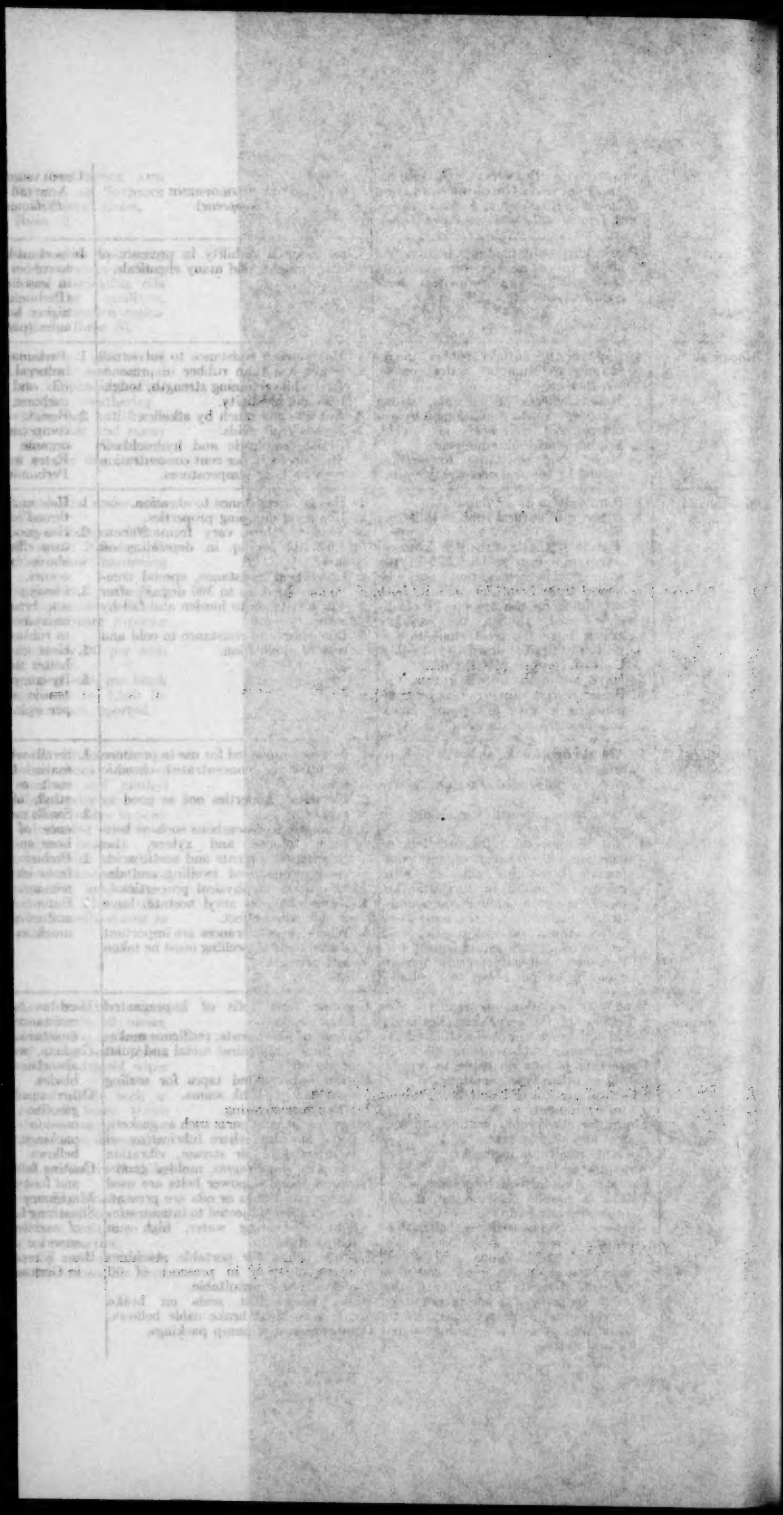
Neoprene and Buna S offer good dielectric strength and a high surface leakage resistance, while Perbunan and Thiokol are less satisfactory as electrical insulators. All of the materials suffer some depreciation at high temperatures.

Chemical Properties: With respect to chemical properties, chief interest centers on the comparative oil resistance, as measured by loss in tensile strength or the amount of swelling. Special grades of Neoprene compounds have been designed to withstand swelling in the presence of oil, though some does take place. A comparison with natural rubber for a Neoprene compound is shown in Figure 3, where the amount of swelling, as measured by the increase of specimen weight, is plotted as a function of time. Com-

Properties of Synthetic Rubber-Like Materials.

	Polyvinyl Alcohol (*Resistoflex*)	Polyalkylene Sulphides (*Thiokol*) (*Perduren*)	Polyisobutylene (*Vistanex*) (*Oppanol*)
General		Characterized by high resistance to a large number of organic solvents.	Inert to most types of deterioration and in one form retains flexibility down to −50 deg. C.
Chemical	1. Resistant to gasolines, oils, greases, kerosene, certain organic solvents such as chlorinated hydrocarbons, aliphatic and aromatic ketones; but not mixtures involving any water, glycerol or alcohol.	1. Resistant to organic solvents, does not swell much in the most active of solvents and very little in presence of oils. 2. Resistant to oxidizing effects of air, ozone, weathering and natural aging. 3. Highly resistant to moisture absorption, permeation and harmful effects of oil and gasoline.	1. Resists most common chemicals, ozone, effects of animal and vegetable oils and fats, moisture. 2. Not soluble in oxygen containing solvents such as alcohols, esters and ketones.
Mechanical	1. Has high tensile strength, good resistance to impact and abrasion. 2. Permeability is about 1/20 that of rubber. 3. Molds readily.	1. Burns slowly. 2. A thin layer sufficient to prevent penetration of gases and liquids. 3. Thiokol DF (or D) has higher physical properties and abrasion resistance than Thiokol A. 4. Thiokol DF has a wide operating temperature range, −40 to 100 deg. C. 5. Thiokol DF is useful for tough heat-and-oil-resistant service. 6. Thiokol A is useful where moisture resistance is required.	1. Has tensile strength and rebound similar to rubber. 2. Has good electrical properties, dielectric constant of 2.2–2.3 and low power factor. 3. Electrical properties do not change much when immersed. 4. Has little or no odor.
Limitations	1. Should not be used in presence of water, alcohol or glycerine because it will dissolve or breakdown. 2. Temperature range limited to characteristics provided by alcohol base, which are necessarily low for flexible grades.	1. Thermoplastic — will reshape under pressure. 2. Thiokol A hardens at subzero temperatures and should not be recommended for use above 150 deg. F. 3. Attacked by strong oxidizing agents and alkalies. 4. Not as high physical properties as rubber but sufficient for many uses. 5. Relatively poor electrical qualities. 6. Cold flow not harmful when restrained as in vibration damper bushings or by fabric as in oil-slinger rings, otherwise it is more than chloroprene or butadiene copolymers.	1. Does not resist chlorine or bromine. 2. Depolymerizes in strong sunlight. 3. Burns freely. 4. Becomes soft when heated to 100 deg. C. for long periods. 5. Soluble or subject to swelling in petroleum and coal tar distillates, chlorinated solvents.
Some Applications	Vacuum and pressure hose where its strength is needed. Handling refrigerants because of its resistance to solvents. Valve seats in carburetors and other equipment where no water is present. Used for conveying solvents, vegetable oils, cleaning fluids. Used for tubing, diaphragms and in other forms for fuel and brake systems, and in hydraulic equipment. Used for oil filter lines, diesel fuel lines. Connections for vibration absorption or isolation.	Airport service hose, airplane fuel hose, has been used for oil lines up to almost 225 deg. F. Gaskets of impregnated fabric, here cold flow makes for good sealing of the joint. One auto company uses with gasket of special carton paper foil for good sealing. Oil-proof belts, hose lines for vacuum power brakes. Used as a flexible non-insulating jacket for electrical cable giving almost as good protection as lead, better where severe moisture or corrosive conditions exist. Also used where vibration exists or for protecting cables of portable tools where oils, solvents or acids plus abrasion exist.	Principal use is in compounding other mixtures, being used for example to improve the abrasion resistance of rubber. Its long fibrous molecules combined with vulcanizing characteristics of other rubberlike materials (polychloroprene or butadiene) are expected to provide new materials of useful properties such as high resistance to abrasion. Combined with rubber good electrical properties result but compound deforms with heat and pressure.

	Plasticized Polyvinyl Chlorides and Polyvinyl Chloride Acetates (*Flamenol, Knightware, Korolac, Koroseal, Irrolite XTE-80 and Nelco Tape*)	Polychloroprene (*Neoprene*)	Copolymerized Butadiene and Acrylic Nitrile or Styrene (*Perbunan, Perbunan Extra, Buna S*)
General	Particularly outstanding because of ability to seal against corrosive acids, have good flexing properties, have good resistance to oxidation.	Chief asset is stability in presence of oils, sunlight, and many chemicals.	Important because of its superiority to rubber in resistance to swelling in gasoline and lubricating oils (Perbunan), aging qualities, higher heat and abrasion resistance (particularly Buna S).
Chemical	1. Superior to natural rubber in resistance to sunlight, water, ozone, weathering. 2. Resists effects of solvents, strong alkalies, caustic solutions, hydrochloric acid, oleic acids, nitric acid, sulphuric acid, chromic acid. 3. Have good resistance to swelling caused by mineral or vegetable oils.	1. Has marked resistance to solvents. 2. Swells less than rubber in presence of oil while retaining strength, toughness and elasticity. 3. Not affected much by alkalies. 4. Resists fruit acids. 5. Resists sulphuric and hydrochloric acids up to 30 per cent concentration even at high temperatures.	1. Perbunan resists effects of mineral, vegetable and animal oils and fats; aliphatic hydrocarbons; sunlight, weathering. 2. Resists swelling by anti-freeze compounds, gasoline and many organic solvents (Perbunan Extra to a greater degree than Perbunan).
Mechanical	1. Burn only in direct flame. 2. Superior to natural rubbers in flexing life. 3. Tensile strength of harder Koroseal compounds may reach 9,000 lb. per sq.in., while elongation may be varied from 2 to 500 percent, both depending on the amount of plasticizer used, though the stronger grades have the least elongation. 4. Will stay flexible down to —20 deg. C. with proper plasticizing. 5. Have low moisture absorption. 6. Resist corona cutting, can be compounded to give low power factor and dielectric constant.	1. Has good resistance to abrasion. 2. Has good damping properties. 3. Tensile values vary from 200 to 4,000 lb. per sq. in. depending on mix. 4. Good heat resistance, special compounds good up to 300 deg. F. after which it tends to harden and fail by embrittlement. 5. Like rubber in resistance to cold and ease of application.	1. Has small cold flow under continued loading. 2. Has good resistance to temperature effects up to 200 deg. F. above which slow hardening occurs. 3. Flexing properties, permanent set, creep, hysteresis loss and resistance to moisture superior to rubber. 4. Heat conductivity 20 per cent better than rubber. 5. By compounding with gas black tensile values up to 4,500 lb. per sq.in. have been reported.
Limitations	1. Are thermoplastic and will reshape under pressure. 2. Not recommended for use above 150 deg. F. 3. Not recommended for flexing in presence of hot oil. 4. Not recommended for complete or constant immersion in organic compounds containing chlorine, nitro groups, aliphatic or aromatic ketones, aromatic amino compounds, lacquer solvents, acetic anhydride. Softer grades will lose plasticizer in certain oils, gasoline and benzol. 5. Recommended only for low tension cable insulation, below 600 volts.	1. Not recommended for use in presence of nitric or concentrated chromic acid. 2. Electrical properties not as good as rubber. 3. Aromatic hydrocarbons such as benzene, toluene and xylene, also chlorinated solvents and acetic acids cause pronounced swelling and deterioration in physical properties. 4. Esters, such as amyl acetate, have deteriorating effect. 5. Where close tolerances are important the amount of swelling must be taken into account.	1. Swells when in contact with aromatic hydrocarbons, ketones such as acetone and methyl ethyl, alcohols, esters. 2. Swells more than rubber in presence of chlorinated hydrocarbons and organic bases. 3. Perbunan is inferior to rubber from electrical standpoint, not recommended for insulation. 4. Buna S in contact with gasoline and mineral oil swells almost as much as rubber.
Some Applications	Used as insulation on electric wire because of its permanent flexibility and resistance to deteriorating effects which make rubber unsuitable. Protecting jackets on cable to replace lead or other type sheathing. Chemical and oil resistant hose, tubing and equipment. Lining for chromium, pickling stainless steel and similar tanks. Covering electroplating racks. Wrapping material. Fabric coating for waterproofness. Tubing to handle chlorine gas, ozone, sulphuric, and hydrochloric acid. Mechanical goods such as belts, diaphragms, machine aprons, seals, gaskets for grease guns and shock absorbers, expansion joints, and as a resilient covering for rolls in the textile industry. It has proper drag for yarns and necessary resistance to acids and oils used in spinning, weaving and dyeing.	Gasoline tank cells of impregnated fabric. Carburetor valve seats; resilience makes for tight seal against metal and quiet operation. Cotton impregnated tapes for sealing float and gas tank seams. Ignition wire covering. Many mechanical parts such as gaskets, belts, air hose where lubricating oil is injected in air stream, vibration dampers, diaphragms, molded goods. Neoprene bonded power belts are used where acid fumes or oils are present. Dredger cables subjected to intense sunlight, oil-bearing water, high sun temperature. Electric cables for portable machines where abrasion in presence of oil makes rubber unsuitable. Brake hose, dust seals on brake cylinders, hand brake cable bellows. Compression and pump packings.	Used as jacket for cables where resistance to solvents, oil, ozone, moisture and abrasion are factors. Gaskets, water pump seals, shock absorber seals, windshield wiper blades. Other mechanical goods such as gasoline and oil hose, transmission and conveyor belts, packings, steam hose, diaphragm, bellows. Coating fabrics, protective clothing and footwear. Machinery mounting blocks. Sheathing for wire and hose because of excellent heat resistance and superior aging properties. Buna S reputedly used extensively in Germany for automobile tires.



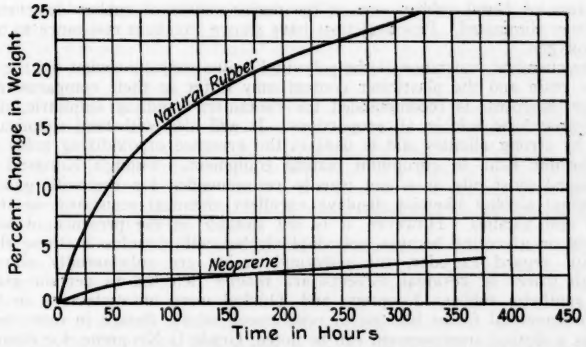


Fig. 3—Comparative resistance of Neoprene and natural rubber to vegetable oil indicates possibilities of properly compounded rubber-like materials for acid resistance

parative ratings are also shown in Table II. It is often pointed out that the small amount of swelling occurring in Neoprene compounds or in Buna rubbers does not much affect their physical properties and consequently they may be as satisfactory in some applications as non-swelling rubber-like materials. However, for a minimum amount of swelling in the presence of most mineral, animal, or vegetable oils, Thiokol or the polyvinyl resins have been more satisfactory.

TABLE II.—CHEMICAL PROPERTIES.

Resistance to Lubricating and Vegetable Oils*

MATERIALS	SWELLING	LOSS IN TENSILE STRENGTH
Natural rubber	large	large
Neoprene	small	slight
Buna S	small — larger	slight
Perbunan	slight	slight
Thiokol	very slight	very slight
Koroseal	none	none
Resistoflex	none	none

* *Resistance decreases markedly with increased temperatures*

Perbunan (7 per cent nitrogen) is best among the Buna rubbers from the standpoint of oil resistance, though its performance is improved slightly by a higher percentage of acrylic-nitrile, Perbunan Extra (9 to 10 per cent nitrogen). Like natural rubber and Neoprene only more slowly, these Buna rubbers will slowly swell and dissolve in chlorinated and aromatic hydrocarbons such as benzol, toluol, and carbontetrachloride.

The Butyl rubbers exhibit unusually good acid resistance, it being reported that they will withstand concentrated sulphuric and nitric acid for a greater length of time than natural rubber. They are, however, swollen by aliphatic hydrocarbon solvents as readily as natural rubber, though they are more resistant to simple aromatics such as benzene and toluene. It has been believed, however, that by eliminating unsaturation in the chemical

structure of Butyl rubber, one of the major causes of rubber deterioration has been eliminated. Practical tests have shown excellent resistance to ozone and oxygen.

The chemical resistance of the polyvinyl resins vary somewhat for the synthetic resin and the plasticizer content may differ in their comparative behavior. Koroseal is recommended for containers handling sulphuric, nitric, or hydrochloric acid in all proportions. It will also withstand chemical attack by strong alkalies and is used in the presence of oxidizing acids such as chromic acid, in chromium plating equipment. Though Koroseal will withstand most oils, it is not usually recommended for use with gasoline. Polyvinyl alcohol likewise displays excellent chemical resistance to strong acids and alkalies. However, it is not suitable in the presence of water, alcohol or glycerine because polyvinyl alcohol will dissolve in these fluids.

With regard to odor, the polyvinyl resins are substantially odorless, though traces of residual solvents are readily detected in certain grades. The synthetic rubbers Neoprene and Thiokol were unsatisfactory in their first commercial forms because of pronounced odors, though in more recent grades a distinct improvement can be noted, Grade G Neoprene, for example, being quite satisfactory in this respect.

Effect of Temperature: At higher temperatures the rubber-like materials will swell to a greater extent in oil; the thermoplastic compositions will creep much more rapidly under load; and generally poorer electrical properties will be displayed. Very low temperatures will stiffen the synthetic rubbers, however, Thiokol will retain its resiliency at lower temperatures than the others. Butyl rubbers compare favorably with natural rubber when aged at 100 degrees C. while Butyl rubber loses little or no tensile strength the natural rubber becomes embrittled.

Among the Buna rubbers, Buna S possesses the best resistance toward high temperatures, suffering the least change upon prolonged exposure. Perbunan and Neoprene will harden slowly upon prolonged exposure to heat. Certain compounds of Neoprene do not begin to harden until temperatures of 300 degrees F. and above are reached.

On the other hand, polyvinyl resins should be used at temperatures below 150 degrees F. though they will not tend to harden above that. While short exposures in excess of this temperature may be feasible in some circumstances, it must be remembered that creep will occur at an accelerating rate when load is applied, with a tendency to deform.

HIGH-PRESSURE BOILER-FEED PUMPS.

This article has been condensed from a paper written by Igor J. Karassik, application engineer, Worthington Pump and Machinery Corporation. Mr. Karassik's paper was awarded first prize in a competition held by the Hydraulic Institute. This extract of his prize paper is reprinted from the March, 1941, issue of Power.

The severity of modern requirements has introduced new problems not only from the point of view of design and construction of boiler-feed pumps, but also from that of application and operation of all the equipment in a steam power plant. Some 20 or 30 years ago, feedwater temperatures in excess of 212 F. and pressures over 250 pounds were rarely encountered. The modern centrifugal boiler-feed pump is called upon to handle feedwater at temperatures anywhere from 200 to 500 F. and pressures up to 3000 pounds, depending on the type of installation and the particular feedwater cycle selected by the power-plant designer.

The latter has a choice between two different types of boiler-feed systems insofar as location of the pump in the feed cycle is concerned. The first is based on pumping the feedwater directly from the deaerating heater up to the full required discharge pressure, the feedwater then passing through closed heaters on its way to the boiler.

The second system involves using two separate boiler-feed pumps operating in series; the closed heaters are between these two pumps. This last system may be chosen in order to (1) reduce the initial cost of the closed heaters, (2) retain existing pumps in the case of a topping unit, (3) meet some special requirements for the slower speed of 1750 RPM., such as the use of wound-rotor motor drive, which would otherwise require as many as 12 stages in a single pump casing.

However, both mechanical and economic advantages result from the use of the single-pump system. A boiler-feed pump, which overcomes the entire head from the deaerating heater to the boiler-feed regulating valve, conserves floor space, reduces the number of stuffing boxes to be packed, eliminates high-pressure and high-temperature stuffing boxes and saves power by handling colder water. This last saving is of an appreciable magnitude, since the power required to deliver a certain weight of water against a given pressure is inversely proportional to the specific gravity of the water. Thus, for instance, a pump handling 400-F. water will consume 12 per cent more power than at 212 F.

In addition, the construction of a centrifugal pump is rendered more complex by the necessity of designing stuffing boxes that will perform satisfactorily under the high suction pressures and temperatures to which they are subjected. A range is ultimately reached which precludes the possibility of packing the stuffing boxes without the use of pressure-reducing labyrinths. These are bled back to some region of lower pressure in the feed cycle in order to reduce the stuffing-box pressure. The amount of feedwater bled back through the labyrinths must be handled by the primary pump in addition to the normal capacity of the system, involving a certain additional expenditure of power.

In order to illustrate the difference in power consumption resulting from the two systems, a typical example is presented in Figure 1. It is assumed that the secondary feed pump of the second system is fitted with pressure-reducing labyrinths, the bleed-off capacity being 3 per cent of the full-load capacity. The water horsepower to be developed by the single pump is 1088, while the 2-pump arrangement must develop 1204 HP., a difference of nearly 11 per cent. Assuming an efficiency of 70 per cent for all the pumps, the second system will require 165 more brake-horsepower at full load.

For a complete evaluation of the operating costs of the two systems, a comparison must be carried out over the full load range, with a reasonably accurate knowledge of operating hours at various loads. In this connection, it is important to bring out the fact that the percentage of power increment of the second system over the first increases at part loads, because the quantity of labyrinth bleed-off is the same.

The wide range of temperatures in which boiler-feed pumps are required to operate makes it imperative to consider the effect of feedwater temperature on the pump construction and on the choice of materials for the various pump parts. This is such an important factor in maintenance costs and ultimate life of the equipment that the proper selection of better materials for severe-service conditions involves relatively little additional cost in comparison with the betterment in operation.

The choice between cast iron, cast steel or forged steel for the pump casing material is basically dictated by mechanical considerations, and specifically by the requirements imposed by the pressure to which the casing is subjected, as well as by considerations of relative resistance to erosion

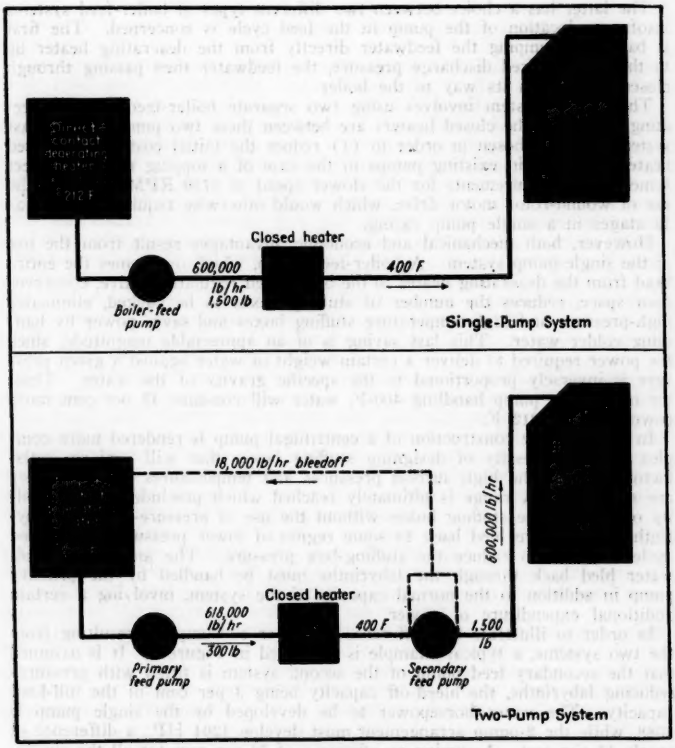


Fig. 1—High-pressure pumping can be accomplished in a single step at low temperature or by a 2-pump setup in which the high-temperature pump may require a leakoff

and lack of porosity. Inasmuch as steel is generally used for pressures over 1000 pounds, temperature conditions need not enter into the choice of material for this range of pressures. For the lower range, however, even though cast iron may prove to be quite satisfactory on the basis of strength alone, cast steel should be seriously considered whenever the temperature of the feedwater exceeds 300 F., and is definitely required for temperatures above 350 F.

CHOICE OF MATERIALS.

In the question of the choice of materials for the boiler-feed-pump fittings, it is safe to say that, if not for all feedwater temperatures, then at least for over 250 F., the use of materials having a coefficient of expansion approximately equal to that of the shaft and of the casing is highly desirable. Thus, from strictly mechanical considerations, steel or stainless-steel fittings

are preferable to bronze, because the coefficient of thermal expansion of bronze is about 40 per cent greater than that of steel. As it is impossible to shrink bronze impellers on a steel shaft, the clearance between the shaft and the impeller hubs will increase as the pump is brought up to operating temperature, introducing the possibility of leakage between shaft and impellers.

The longitudinal expansion of the rotor assembly has an even more marked effect on the condition of the pump. Inasmuch as the bronze assembly will expand to a greater degree than the steel shaft, the impeller hubs will be "crushed" to the extent of the difference in expansion. Since this cannot take place uniformly and evenly, severe moments will be exerted on the shaft, with consequent bowing of the latter. These possibilities are obviously removed when the pump fittings are such that inequality of expansion between the shaft and the rotor assembly does not exist.

For operation at high temperatures, the pump casing should be supported as near to the horizontal centerline as possible. Such a construction will prevent excessive strains due to temperature differentials which might otherwise seriously disturb the alignment of the unit. Certain pump manufacturers, as a matter of fact, have standardized on centerline casing support for boiler-feed pumps, regardless of the operating temperature.

Stuffing-box design is likewise intimately connected with the temperature of the feedwater. Unfortunately, the art of stuffing-box design has not yet reached the status of an exact mathematical science, but is based on empirical knowledge, and therefore differs greatly with the various pump manufacturers. A store of valuable data is slowly being gathered, however, and there is hope of a more rational analysis of stuffing-box performance in the near future.

Experience has shown that adverse conditions accumulate until a certain combination of pressure, temperature and other adjunct factors is reached when a stuffing box can no longer be packed successfully. It then becomes necessary to incorporate pressure-reducing labyrinths ahead of the stuffing box proper. It is of course impossible to set exact limiting conditions for the use of these labyrinths, as they depend on the diameter of the boxes, the operating speed and the relative excellence of the cooling system, as well as on the prevailing pressure and temperature.

However, we know from experience that, with proper cooling, it is possible to pack satisfactorily a 4-inch stuffing box operating at 3600 RPM., when subjected to 400-pound pressure at 400 F., provided the pump is perfectly balanced and operates without vibration at any load. Operation of a centrifugal pump at abnormally light flows, as occurs frequently in some stations, increases shaft vibration and is therefore detrimental to good stuffing-box performance.

Figure 2 illustrates qualitatively the effect of temperature and pressure on a factor which is assumed to represent stuffing-box wear. No values are given, as the curves are indicative of the relation only, and are intended more as a suggestion for the recording of experimental results than as an actual description of facts already determined.

While the minimum net positive suction head (in excess of vapor pressure) should theoretically be independent of the feedwater temperature, the latter has an important bearing on satisfactory suction conditions in practice. When a boiler-feed pump takes its suction from a direct-contact heater at a relatively high temperature, it is recommended that a safety factor be provided in establishing the required submergence.

This safety margin compensates in part for possible fluctuations in the heater pressures and for the time lag between a sudden reduction of pressure in the heater and the ultimate reduction of temperature at the pump suction. This effect is more accentuated in the higher temperature range

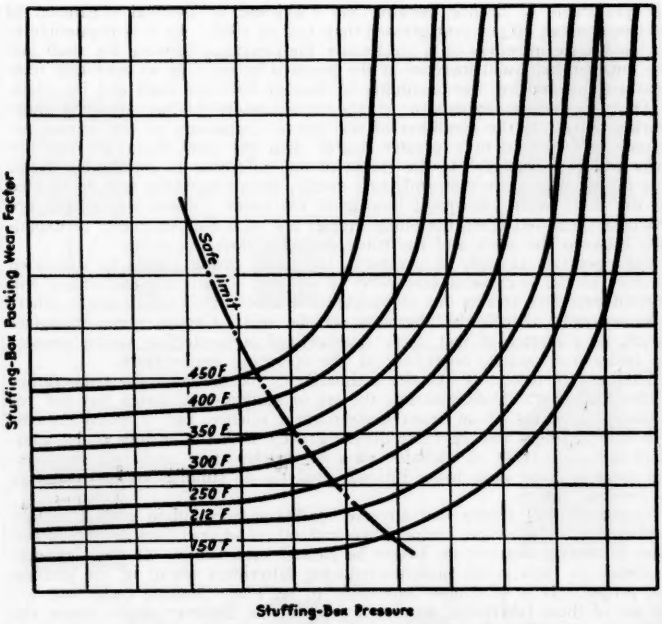


Fig. 2—Pressure and temperature affect the safe limits of stuffing-box operation

because of the more rapid increase in vapor pressure for a given increase in temperature. Thus, for example, a 12-foot differential between the suction pressure and the vapor pressure corresponds to a margin over flashing of 15 degrees at 212 F., and of only 1.6 degrees at 400 F.

ABOVE 300 F.

However, a boiler-feed pump operating at temperatures above 300 F. does not usually take its suction from a direct-contact heater, but is supplied by some type of primary, or booster, pump. As a result, the suction pressure of the main feed pump corresponds to the discharge pressure of the primary pump (less line losses) and no difficulty is experienced in providing a margin from 50 to 200 pounds over the vapor pressure.

In general, while feedwater temperature should be given all due attention in analyzing the suction conditions, the desire to provide proper and safe operating conditions for the boiler-feed pump should not lead to the imposition of unnecessary and impractical limitations on associated equipment of the feedwater system.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

FACTORS AFFECTING WELDING COSTS.—This article by C. M. Taylor, Vice-President of the Lincoln Electric Company, illustrates how labor, electrode, and power costs may be reduced to a minimum by a detailed study of electric arc welding operations. The article is reprinted from the December, 1940, issue of Machinery.

The factors comprising direct welding costs are labor, electrodes, and power. The labor cost depends for a particular set-up on how many pounds of effective metal are deposited per hour, or, in other words, how many minutes per hour the arc is in operation. Expressed in per cent, this time is known as the operating factor. Assuming a labor rate of $1 per hour and an arc speed of 50 feet per hour (arc speed is the actual rate of travel of the arc and may be expressed in inches per minute or feet per hour), Figure 1 shows that maintaining a maximum operating factor assures the greatest cost reduction.

The use of proper jigs and fixtures, and correct set-up, aids in keeping the operating factor high. As an example, assume that it requires a welder two minutes to weld a job, and two minutes, or slightly less, to set it up. The time is, then, four minutes per part; or the production is 15 per hour. By providing a helper and another jig, the helper can set up while the welder is welding; the production is increased to 30 units per hour, reducing the cost materially. A production of 15 parts per hour requires a jig, a welding machine and a welder, whereas a production of 30 per hour requires two jigs, a welding machine, one helper, and a welder. The cost reduction is obvious, because the second 15 parts are produced at the cost of one jig and one helper.

Also affecting the operating factor is the matter of working position. The work should be positioned so that it is easy and convenient to weld. For example, suppose that the operator can always weld in the "down hand" position. The speed will then be, say, 26 feet per hour. If, however, it is necessary to weld in the "fillet" position, the speed may be only 11 feet per hour.

Assuming that the most favorable set-up has been devised (and this factor is kept fixed), note the effect of changing arc time by means of electrodes. Assume four arc speeds (20, 25, 30, and 40 feet per hour) for different sizes or types of electrodes, the set-up time being the same (2 hours) in each case, to produce a unit of, say, 100 feet of welding. The arc time is 5, 4, 3.3, and 2.5 hours, respectively, for the speeds given; the total time 7, 6, 5.3, and 4.5 hours; the cost (at $2 per hour) $14, $12, $10.60, and $9. There is a marked increase in production (17, 32, and 55 per cent, respectively) with no increase in equipment or floor space. These production increases account for cost reductions of $2, $3.40, and $5, respectively. The factors are plotted in Figure 2.

EFFECT OF ELECTRODE SIZE ON COSTS.

Consider the effect of electrode size on the cost of deposited metal. Assume that all factors, including deposition efficiency (ratio of amount of deposited metal to total electrode consumed), type of joint, and all other items, except the electrode size, are fixed. Labor is $1 per hour; power is $0.02 per Kw. hour; operating factor (ratio of arc time to total time) is 50 per cent; overhead is 100 per cent; and deposition efficiency is 66 2/3 per cent. Comparing 1/8-inch and 3/8-inch sizes, there is a difference of $2.011 in cost per pound deposited, the cost for the 1/8-inch size being $2.564, and for the 3/8-inch size, $0.553. See Figure 3.

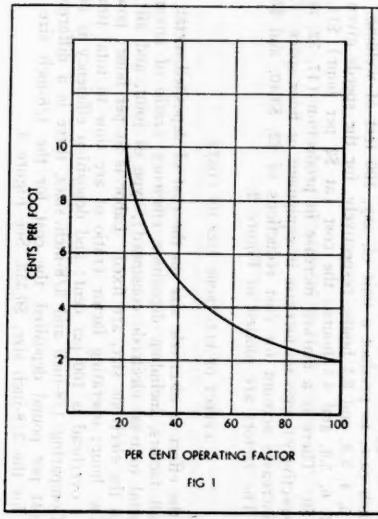


Fig. 1. Chart Showing Effect of Operating Factor on Costs

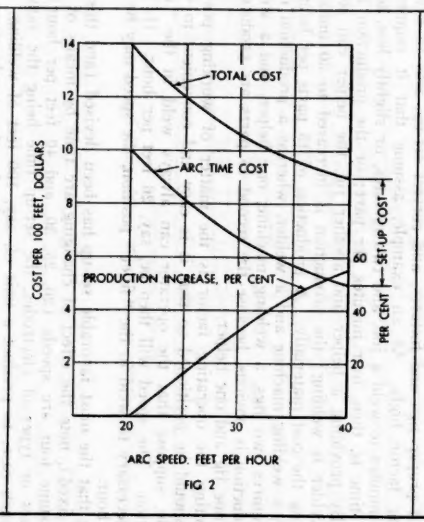


Fig. 2. Chart Showing Effect of Arc Speed on Costs

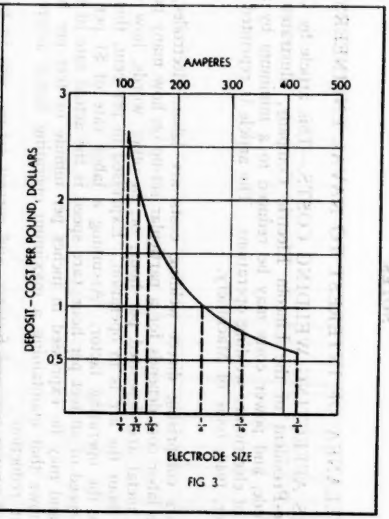


Fig. 3. Effect of Current and Electrode Size on Costs

Cost reduction is not the only benefit obtainable by using the largest size electrode practicable. An improvement in the quality of welding is often assured, especially where multiple passes are used. The distortion of the joint is less with the smaller number of passes required in using larger electrodes.

Controlling stub-end losses, which are part of the electrode cost, effectively reduces costs and is easily accomplished. The cost per pound deposited for stub-end losses of 2, 4, 6, and 8 inches in a 1/4-inch electrode, 14 inches long, is $0.996, $1.030, $1.089, and $1.205; in a 1/4-inch electrode, 18 inches long, $0.958, $0.986, $1.032, and $1.076.

The labor cost per pound deposited increases with larger stub-end losses, due to the greater number of interruptions and the lower operating factor, and the pounds of electrode per pound of deposit increase due to the stub-end waste. On the average, labor costs increase 3 per cent for each 2 inches above the minimum 2-inch stub-end loss.

HOW TO REDUCE POWER COSTS.

Power costs depend on the generator, its efficiency and size, and to some extent on the operating factor. In selecting a welding generator, the most modern design available should be secured, in order to minimize costs. A comparison between a new 40-volt and an old 25-volt machine—the basis being a weld 896 inches long—showed a difference of 51.3 minutes welding time, 2.5 Kw. hour input per weld, and 1.1 inches of weld per electrode in favor of the new unit.

Such operations as attaching of nameplate, painting and sand-blasting, stress-relieving, and heat-treating, not only enter into the total cost of the product as items of cost, but are also to be considered in connection with the welding. The name-plate must be easily placed. Painting involves accessibility to surfaces to be painted; the same applies to sand-blasting.

Special consideration in preparation and welding may be required for stress-relieving and heat-treatment, so as to avoid collapse or deformation when the metal is soft, and provide support for the escape of gases. Parts should not be completely enclosed, since expansion of the air will cause deformation.

Based on the foregoing suggestions for cost reduction, the following points should be considered: Use of jigs and fixtures improves the operating factor and lowers the unit production costs; work-positioners increase the welding speed and reduce the labor cost; the use of the largest size electrode that is practicable minimizes the cost per foot of weld; care in the selection of electrodes assures the best possible performance economy; the most modern, efficient generator possible reduces power costs and increases speeds; easy accessibility for painting, sand-blasting, and attaching of name-plate minimizes finishing costs.

FLOW-INDICATOR FOR LIQUIDS.—The following article, concerning a flow-indicator containing only one moving part, is reprinted from the January, 1941, issue of The Shipbuilder and Marine Engine-Builder.

In a great many instances in general engineering, but more particularly in marine engineering, there is necessity to have flow-indicators, or similar devices, fitted in those pipelines through which continuous circulation is essential. In marine installations, such circuits to which flow-indicators or "no-flow" alarms are an important addition in aiding surveillance of the plant include:—

(1) Forced-lubrication of individual bearings;
(2) Diesel-engine cooling-oil or cooling-water circuits to individual cylinders and pistols;
(3) Cooling water to air-compressor jackets and intercoolers;
(4) Brine-circulation systems of refrigerating installations;
(5) Hydraulic couplings for Bauer-Wach exhaust-turbine sets;
(6) Supply of lubricating oil to sprayers for main reduction gearing.

To meet the demand for fluid-flow indicators many types of both visual and audible alarm systems, sight-glasses, etc., have been developed in the past. Certain of these entail fairly complicated mechanism. Sight-glasses, particularly with oil, are liable to cloud over, while open tundishes for both hot oil and water have disadvantages. Interest, therefore, attaches to the "Arkon" liquid-flow indicator, which is manufactured by Messrs. Walker, Crosweller & Co., Ltd., of Cheltenham. This device provides a simple, yet reliable, means of visual indication of flow, and can be installed in either the actual pipeline or in parallel in a small-bore by-pass pipe.

A photographic reproduction of an Arkon indicator is given in Figure 1. The body is cast in gunmetal or bronze, and is screwed at both ends for connection in the pipework. Flow through the body is guided so as to rotate a small pressed bronze propeller, which is itself attached to a stainless-steel rod. This rod, which is mounted in a ball-bearing at its lower end, protrudes through the top of the body and carries a disc, painted red on one side and black on the other. This disc is encased in a glass dome, which is held in position on the body by a liquid-sealing ring and a brass ferrule. Figure 2 shows a sectional view with details of the interior.

The spinning disc can be observed at long range, and thus the engineer in charge can rapidly check over the functioning of his essential circuits. By suitably adjusting the propeller, the device can be arranged for either low or high velocity of flow, or for use in oil of any viscosity.

The design of indicator is suitable for pressures up to 50 pounds, but the glass dome is tested to 10 times the working pressure. The range of standard sizes in which the device is available is ½, ¾, 1, 1¼, 1½ and 2-inch diameter, but additional sizes can be specially made, or alternatively, in larger diameter pipelines a standard fitting in a smaller bore by-pass pipe can be used. The dimensions of the ½-inch diameter size are 3½ inches between faces and 3⅝ inches from center-line of pipe to glass dome, while the corresponding figures for the 2-inch diameter fitting are 8 inches and 4⅝ inches respectively. In regard to pressure drop through the indicator, this is small. For example, the drop is about ¾ pound per square inch when passing 200 gallons per minute through the ¾-inch diameter fitting, and 3½ pounds per square inch for 800 gallons per minute through the 2-inch diameter fitting.

CAUSES OF DIESEL ENGINE BREAKDOWNS.—This brief abstract of a prize winning paper submitted to the Edison Electric Institute is reprinted from the February 1941, issue of British Motor Ship.

The Edison Electric Institute, which carries out a large amount of valuable research work in America, offered a prize for a paper dealing with the causes of breakdowns in Diesel engines. The successful author gave some notable details in which he analyzed data concerning failures or minor mishaps of Diesel machinery over a period of five years, and his investigation covered the operation of well over 200 oil engines. Including accidents,

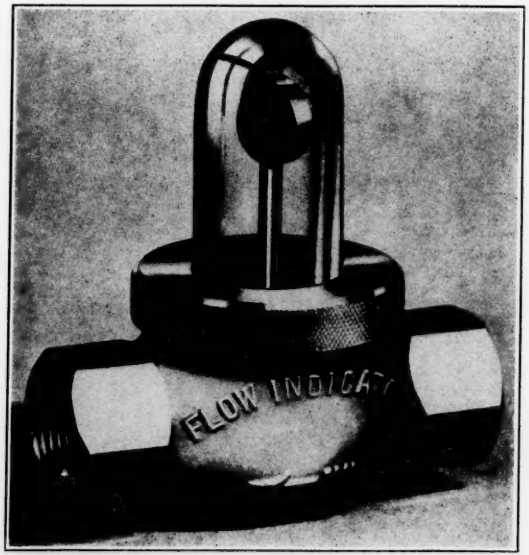


FIGURE 1.—LIQUID FLOW-INDICATOR FOR PIPELINES.

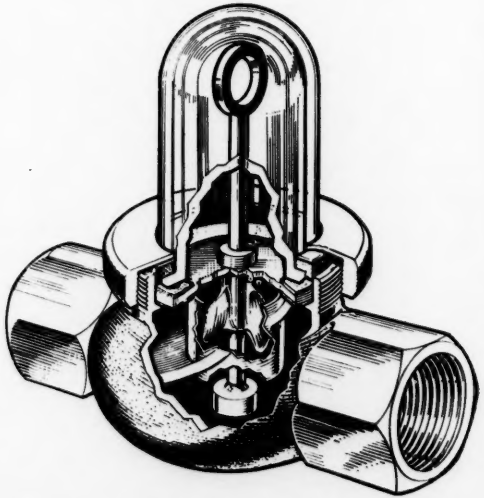


FIGURE 2.—SECTIONAL VIEW, WITH DETAILS OF THE INTERIOR.

overhaul, inspection, adjustment, wear and tear, it was found that there were 31 different causes for engines being out of service and the table gives the more important of them.

We gather that the majority of these engines are of the stationary type, and possibly in marine work there would be some difference in the causes. Evidently the examination, repair and replacement of pistons and piston rings represent the main cause of engine stoppage, apart from inspection and adjustment, and air compressors appear to be subject to failure to a considerable extent. This would not enter with more modern engines, all of which are now of the airless-injection type. Crankshaft troubles appear to be surprisingly large in respect of total hours out of operation, but the number of engines affected is relatively small, and it is obvious that in each case it is a question of the plant affected being out of operation for a very long period in each case.

Engine part or other cause of Failure or Stoppage	Total No. of Hours of Stoppage in 5 years	Per cent. of Hours of Stoppage in 5 years	No. of Machines Affected
(1) Inspection and adjustment	21,043	24.0	127
(2) Main pistons and piston rings	11,562	13.2	139
(3) General overhaul	10,911	12.4	15
(4) Crankshaft	7,368	8.4	10
(5) Air compressors (air-inj. engines)	4,976	5.7	67
(6) Cylinder liners	3,710	4.2	28
(7) Crank and piston-pin bearings	3,704	4.2	91
(8) Exhaust valves (four-stroke eng.)	2,049	2.3	70
(9) Main bearings (incl. thrust and outboard)	1,967	2.2	54
(10) Fuel-spray valves (priming valves, etc.)	1,807	2.0	87
(11) Cylinder heads	1,653	1.9	60
(12) Governors and governor drives	1,540	1.8	41
(13) Water cooling (incl. scale trouble)	1,447	1.7	66
(14) Air inlet valves	1,401	1.6	74
(15) Fuel pumps, distributors, drives, etc.	1,273	1.4	90
(16) Lubricating-oil system ..	1,230	1.4	71
(17) Fuel-oil system	1,044	1.2	38
(18) Cracked frames and cylinders	991	1.1	8
(19) Crosshead pins and bearings	964	1.1	15

It is noteworthy that crosshead pins and bearings are singularly free from serious trouble. This is perhaps all the more remarkable as it may be assumed that the majority of the engines to which the statistics refer are of the trunk piston type and no doubt run at a relatively high speed.

EXPENSIVE BORING REPLACED BY GROUTED LINERS.—This article by Herbert Chase is reprinted from the Metallurgical Engineering Shop Notes of the January, 1941, issue of Metals and Alloys.

When a large sand casting must have one or more fairly large cylindrical holes precisely parallel or in some other exact relation, the usual practice is to core the holes as near to size as conditions permit and still allow for subsequent machining. The casting must then be set up in a boring mill or some equivalent tool capable of machining the holes with required accuracy. This necessitates rather expensive equipment as well as the time of at least one competent machinist and sometimes, in addition, considerable work by the tool room in preparing the necessary tools and keeping them in satisfactory condition.

At least one large maker of machine tools has developed an effective alternative with a noteworthy saving in cost. An important feature is that no boring mill is required, although some lathe work is done and, if extreme precision is needed, an internal grinder may be necessary. Cuts, however, can be far lighter with this alternative method than are usually necessary, on large castings, especially if tubing close to the required hole dimensions is available.

The alternative to boring to finished size consists in using a machined tube or sleeve to form a liner for the required hole, the tube being first machined (at least inside) to provide a bore of required accuracy. The casting is provided with cored holes to receive the liner or liners; the holes are cored to a diameter a half inch or more larger in diameter than the *outside* of the liner, after which the liner is "grouted" in place.

To do this, the liner or liners must be held in the correct relative position and this necessitates a jig, which, however, need not be very expensive. Rings or some equivalent must be provided to prevent the grouting material from running to waste around the ends of the liner, but this is a simple and inexpensive precaution.

It is important, however, to use a grouting metal that will not contract when it solidifies—which, in fact, will preferably expand slightly when it becomes solid. Otherwise the grout will be loose in the hole and may permit of enough play to destroy the required accuracy of alignment. In expanding when it solidifies, the grout, of course, anchors the liner securely, especially if the liner be left somewhat rough on its outer diameter.

In addition, the holes should be cored in the casting with sufficient accuracy to avoid any appreciable variation in the thickness of grout. If the grout is much thicker on one side of a liner than on the other, its solidification will force the liner slightly away from the thick side, so that the axis of the liner will not be precisely where intended.

In using this process, this particular machine tool maker employs an alloy containing 50 per cent metallic bismuth, 25 per cent tin and 25 per cent lead. This alloy expands when it freezes—a property imparted by the bismuth. Other alloys containing bismuth in sufficient quantity might also be used, but that mentioned has proved satisfactory. The low melting temperature (about 200 degrees F.) of the grouting alloy is also an advantage; it should be remembered, though, that the method can be used only when the temperature reached by the liner in service is below that at which the grouting alloy becomes soft enough to flow. When the precautions mentioned are observed, the results obtained (in a machine in which the liners used must be parallel and spaced within close limits) are reported entirely satisfactory, and the method is economical.

In this instance, the liners position plain bearings, but the utility of the practice outlined might well be extended to include liners for other purposes. It doubtless could be used also to grout in place solid parts, such as ground shafts that must extend from a casting to form an accurately machined mounting for some other part of a machine.

ASSOCIATION NOTES.

The annual banquet of the Society held in Washington on April 25, 1941, was outstanding in a long series of successful events of this kind. It was by far the most largely attended of any banquet which has been held. Reservations were received for 1,228 covers which necessitated a rearrangement of seating facilities in the main ballroom of the Willard Hotel and the use of tables for approximately 100 in the small room adjoining.

A feature of the attendance was the presence of an unusually large number of executives of prominent organizations in the shipbuilding and allied marine engineering industries. The Society was honored by the presence of five past Presidents and a large number of past Secretaries, and also by the attendance of several Chiefs of Bureaus.

The speakers of the evening were the Honorable Frank Knox, Secretary of the Navy, and the President of the Society, Captain Claud A. Jones, U. S. N. Both made stirring addresses of particular interest in view of the present national and international situations. The speakers were introduced by the Toastmaster, Rear Admiral Emory S. Land, U. S. N. (Retired), the Chairman of the U. S. Maritime Commission, in a delightful manner.

Music by an orchestra from the United States Navy Band and the Glee Club, conducted by Lieut. Charles Benter, U. S. N., contributed materially to the enjoyment of the evening.

The President announced that Mr. Arthur G. Fessenden had been elected a Life Member of the Society by the Council and awarded a cash bonus in consequence of his services as clerk for the Society for the past third of a century.

SOCIETY LAPEL BUTTONS.

The order blank for lapel buttons is republished in this issue, immediately following these notes. These buttons are being sold to members for fifty cents each, postpaid.

MEMBERSHIP.

The following have joined the Society since the publication of the February 1941 issue:

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Agens, F. F., Lieutenant, U. S. Navy.

Atwood, Lewis J., Lieutenant, U. S. N. R., U. S. S. *Pocomoke.*

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ERRATA

The authors have submitted the following substitute page for page 61, the JOURNAL for February 1939, Vol. 51, Number 1.

SILVER BRAZING ALLOYS IN THE MARINE FIELD.

FORMULA

X = (unknown) = depth of shear
D = shear diameter
T = Tensile strength of weakest member
W = Wall thickness of weakest member
Y = Factor of Safety
L = Shear strength of silver brazing alloy

$$X = \frac{W(D-W)\ YT}{L\ D}$$

EXAMPLE

Length of shear depth necessary when joining 3/4 — 0.064 wall thickness copper tube to 3/4 steel tube sheets.

X = unknown
D = 0.75
T = 33,000 annealed value copper
W = 0.064
Y = 5
L = 25,000 psi shear value of silver brazing alloy. (Note: This is an arbitrary minimum value which can be expected from all silver brazing alloys containing appreciable amounts of silver.)

$$X = \frac{.064 \times (0.75 - 0.064) \times 5 \times 33{,}000}{25{,}000 \times 0.75} = 0.386$$

depth of shear

A similar method can be used with flats. In this case the formula becomes $X = \frac{YTW}{L}$

Example: Length of shear necessary when joining 0.050 inch annealed monel metal sheet to a stronger material, or material of equal strength.